DESIGN OF A 15,000 H. P. HYDRO-ELECTRIC DEVELOPMENT ON EAST CANADA CREEK, N. Y.

BY

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ARMOUR INSTITUTE OF TECHNOLOGY

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DESIGN OF A 15,000 H.P. HYDRO-ELECTRIC DEVELOPMENT ON EAST CANADA CREEK, NEAR EAST CREEK, N.Y.

A THESIS

PRESENTED BY

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PREFACE

In the following design the authors have attempted to cover the technical features of a hydro-electric development. They realized their inability to make decisions which would require the judgment of an experienced engineer, and have therefore emphasized the mathematical and physical treatment of the factors which enter into the design, rather than the economic and commercial factors.

Free use has been made of the material on scroll cases and draft tubes written by Hillberg and published in Volumn 72 of the Engineering Record.

We wish at this time to thank the Faculty of Armour Institute of Technology, and in particular Professor Stanley Dean, of the Civil Engineering Department, for their guidance and assistance during the past four years.

27197

Acknowledgment is made to Vielé, Blackwell and Buck of New York for topographical maps furnished and to the Wellman-Seaver-Morgan Co., for data on turbine runners.

May 31, 1917.

J.W.T.
H.A.K.
E.H.S.

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PART 1.

DESIGN OF
DAM, SPILLWAY, PIPE LINE,
PENSTOCKS AND SURGE TANK.

J. W. TIERNEY.

DESIGN OF A 15000 H.P. HYDRO-ELECTRIC DEVELOPMENT ON EAST CANADA CREEK, NEAR EAST CREEK, N.Y.

The proposed development is located on the East Canada Creek, about a mile and one-half above it's junction with the Mohawk River. This development supersedes an old hydro-electric plant now in operation at the same site. The company controlling the water rights for this development is also the owner of a hydro-electric plant on the same river at Inghams Mills, three and one-half miles above the site under discussion. The two plants will be operated in parallel.

Due to topographical conditions the storage area available at a reasonable cost is negligible, and the total storage supply would be consumed in three days relying on storage alone. Therefore to tide over periods

of low water an auxiliary steam plant will be required, which however, is not considered in this thesis which is specializing on the hydraulic features only. The subjects which will be discussed here are the dam, pipe line and it's accesories, the turbines and their accesories and the power house with the electrical equipment.

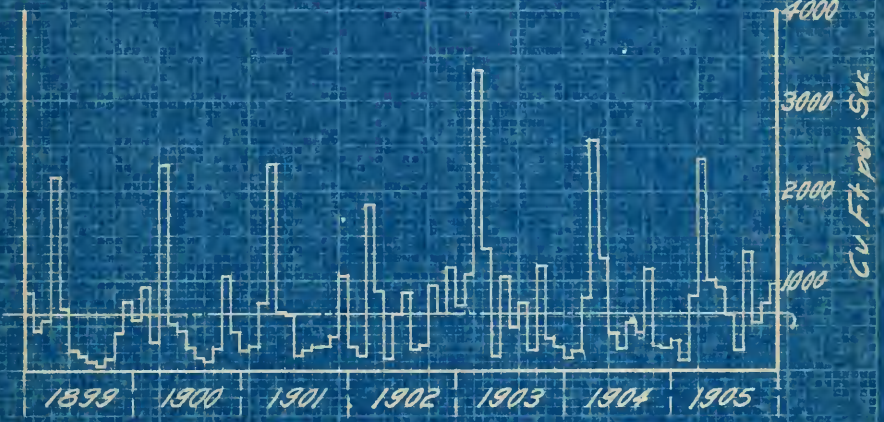
Incorporated in the drawings are several features which we believe are original.

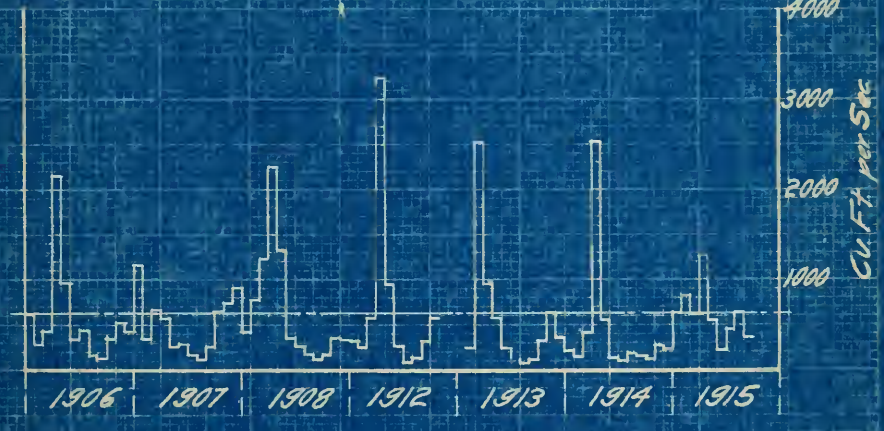
On the pages immediately following are tables showing the average monthly flow of the river for the last fiften years.

Fig. #1

Mean Monthly Flow Curve

Based on readings taken on East Canada Creek
at Dolgesville, N.Y. and corrected for difference
in area between Dolgesville and the dam site
Area at Dolgesville = 256 sq. mi. At dam = 281 sq. mi.
Dashed line indicates average of monthly flows





Mean monthly flow of East Canada Creek at the dam site in cubic feet per second, based on readings taken on East Canada Creek at Dodgeville, N.Y., and corrected for the difference in drainage area between Dodgeville and the dam site.

MONTH	1899	1900	1901	1902	1903	1904
Jan.	896	585	258	283	735	185
Feb.	483	976	273	196	1070	238
Mar.	570	320	760	1860	3340	832
Apr.	2170	229	2300	895	1355	2470
May	696	535	692	111	171	1220
June	215	407	620	622	1010	423
July	182	243	191	885	492	274
Aug.	106	158	209	214	764	541
Sept.	101	146	274	268	205	435
Oct.	123	112	277	950	1170	1150
Nov.	415	1050	395	645	349	279
Dec.	776	405	1030	1155	283	271

(continued)

Table continued.

MONTH	1905	1906	1907	1908	1909	1910
Jan.	338	620	1190	442	455	306
Feb.	127	299	377	795	567	242
Mar.	840	415	695	1230	2350	558
Apr.	2390	2160	580	2280	1500	3220
May	1000	990	270	1330	854	915
June	915	354	286	345	630	258
July	635	422	175	290	199	112
August	248	193	152	188	235	120
Sept.	1310	131	260	137	341	300
Oct.	563	352	670	152	262	585
Nov.	765	505	730	339	370	---
Dec.	990	444	900	327	206	---

MONTH	1913	1914	1915
Jan.	---	226	635
Feb.	258	185	865
Mar.	2500	441	640
Apr.	920	2520	1290
May	595	543	560
June	263	168	234

(concluded on next page)

MONTH	1913	1914	1915
July	142	144	495
Aug.	102	192	667
Sept.	122	183	380
Oct.	356	154	---
Nov.	685	293	---
Dec.	387	283	---

LOAD.

This development is made for the purpose of supplying energy to an electric traction system, house and street lighting, manufacturing, and for a typical central station load. Since no definite data is available as to the relative magnitude of the different loads listed above, the average load curve of the Commonwealth Edison Co., of Chicago for the year 1905 is adopted as the typical load curve of this development. The loads are similiar, both combining railway, central station, manufacturing and lighting loads. To determine the most economical pipe line

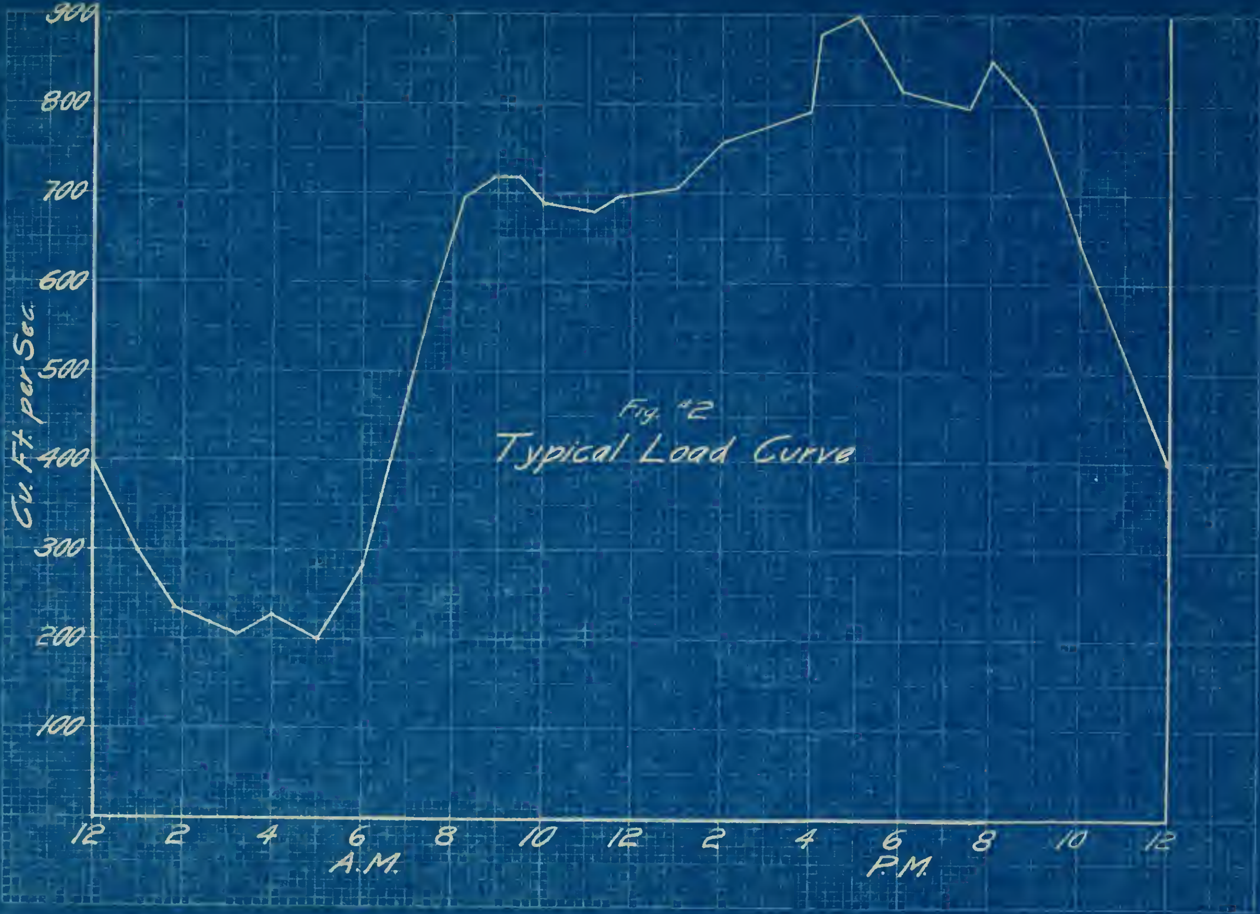


Fig. #2
Typical Load Curve

size it is necessary to know the average and root mean square of the value of the flow through the line. A tabulation of the values taken from the curve is listed below.

TIME	FLOW CUSECS.	$(\text{FLOW})^2$
A.M. 1	300	90000
2	240	57500
3	210	44000
4	230	53000
5	200	40000
6	280	78000
7	490	240000
8	670	450000
9	720	520000
10	690	475000
11	680	462000
12	700	490000
P.M. 1	710	505000
2	760	580000
3	780	610000
4	790	625000

TIME	FLOW CUSECS.	(FLOW)^2
P.M. 5	900	810000
6	820	670000
7	810	660000
8	850	720000
9	800	640000
10	423	520000
11	520	270000
12	400	160000
	14180	9672500

From which the average flow = 590 cu. ft. per sec., and the square root of the mean square of the flow = 635 cu. ft. per sec.

INCOME.

It is necessary in order to be able to calculate whether the plant will be a paying proposition or not to know that the income per year per foot head is. In the following calculations we find that it is $ 1750.00 per year per foot head, which gives an idea of the income of the plant.

The average flow = 590 cusecs., under a head of 180 ft. The overall efficiency is 40% and the net income per K.W. hour is one cent. The horse power output is

$$H.P. = \frac{590x62.5x180x0.40}{550} = 4820 \text{ H.P.}$$

which is equal to 3600 K.W.

The net income per hour is

$$3600x0.01 = \$ \ 36.00$$

and the net income per year is

$$365x24x36 = \$ \ 315,720$$

Therefore "i", the income per foot head per year is

$$"i" = \frac{315720}{180} = \$ \ 1750.00$$

DAM.

Three possible types of dams suggest themselves for this project; reinforced concrete, gravity concrete and earth fill. Although no detailed estimate of the relative costs was made it seems quite evident that the latter type is the more economical due to the hills on both sides of the dam,

from which the earth fill can be placed, eliminating costly quarrying, mixing and placing, and the large amount of skilled labor.

The high water elevation in the forebay is fixed at EL. 515. This is the maximum economical head, entailing a minimum flooding of the property bordering upon the stream. The spillway is designed to carry off the maximum recorded flood flow, therefore it is reasonably safe to assume that the water will never rise above EL. 515. The top of the dam is brought up to elevation 525., to insure against overflowing at the dam section. The data of the dam is as follows:

Maximum height = 70' - 00"

Upstream slope = 3 to 1.

Downstream slope = 2 to 1

Core wall = 2' - 00" wide at the top and is battered 1 in 40

Upstream face of the dam is rip-rapped 16" deep on top of 6" of gravel fill. The dam is founded upon solid rock.

SPILLWAY.

To safeguard the earth fill dam against flood flows it is necessary to provide a gravity spillway. The maximum recorded flood flow is 8732 cusecs., and due to the equalizing action of the hydro-electric plant upstream it is not probable that the flow will ever exceed this value which was recorded before the upstream plant was put into service. To provide against excessive flood the spillway will be designed to carry off 10,000 cusecs.

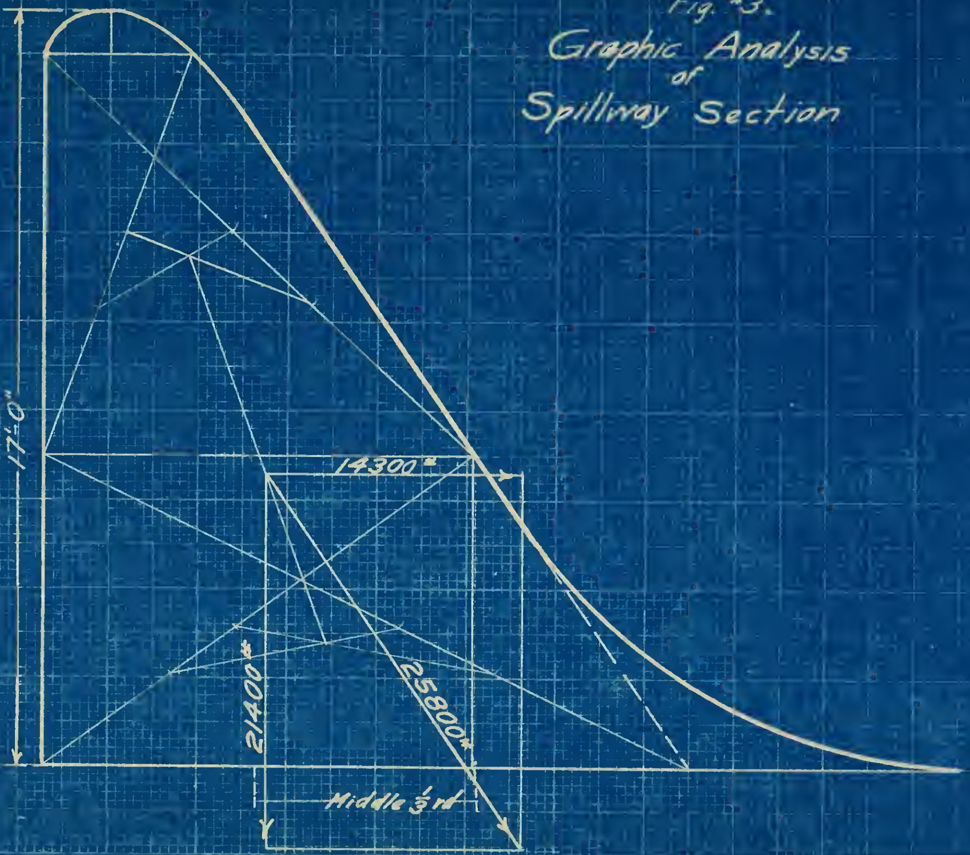
The data for the calculation of the spillway is given as follows:

Q = flow over spillway in cusecs. is 10,000

h = head of water over crest = 5 ft.

L = length of spillway = 262.5 ft.

Fig. #3.
Graphic Analysis
of
Spillway Section



The value of h = 5 ft. which was stated in the above data was gotten by substituting in the weir formula as follows:

$$Q = 3.33\ Lh^{3/2} = 10000 = 3.33 \times 262 \times h^{3/2}$$

from which h = 5 ft.

The downstream face of the spillway must conform to the shape of the curve of water flowing freely over the crest. This curve is a parabola. In a time t the water travels horizontally a distance $x = vt$, v being the velocity over the crest in feet per second. The vertical height through which the water has fallen in the same time is $y = \frac{gt^2}{2}$. The following table gives the values of x and y for the parabola, being calculated from the above equations.

x	1.000	2.000	3.000	4.000
y	0.278	1.112	2.432	4.440

The stability against overturning is determined by a graphical analysis of a section one foot in length, as follows:

Height of maximum section = 17 ft.

Head of water over crest = 5 ft.

Total head of water = 22 ft.

Weight of water per cu. ft. = 62.5 lbs.= w

P = total horizontal pressure = $\frac{wh^2}{2}$

P = $\frac{62.5(22)^2}{2}$ = 14300 lbs., acting at 6.68 feet above the base.

W = total vertical force acting on the section, which is the weight of water per foot of spillway section. Section area = 144.25

W = 62.5x144.25 = 21400lbs.

Thus by combining these two forces vectorially the resultant is found to be 25800 lbs., acting inside of the middle third of the base.

Since the maximum pressure on the base, which occurs at the intersection of the downstream face and the ground is equal to twice the average distributed pressure (the design being based on the middle third principle) the maximum crushing unit

stress is

$$S = \frac{2x21400}{20.6} = 2,080 \text{ lbs., well within}$$

the safe crushing strength of rock.

To guard against the dam sliding downstream, the base is intrenched as shown in the drawings.

PIPE LINE.

In this development in which a pipe line 6445 ft., long is used, the cost of the line enters into the total cost as an appreciable percentage. It is therefore imperative that the most economical pipe be used. The formula used in this connection was derived by M. M. Warren, and was published in the " A. S. C. E. " Proceedings, October, 1914.

Nomenclature.

B = cost of pipe in place, per ft., in diameter, per foot in length.

C = coefficient in Chezy formula for friction in pipes.

D = inside diameter of pipe in feet.

i = income in dollars per year, per foot of head.

L = length of pipe in feet.

p = percentage return on investment.

q = root mean square of flow thru penstock in cusecs.

V = velocity of water in pipe in feet per second.

R = hydraulic radius in feet.

By the Chezy formula the head in the pipe is

$$\frac{LV^2}{C^2R} \text{ feet in which } R = \frac{D}{4}$$

The income lost on this head is

$$\frac{LV^2}{C^2R}\, i \text{ dollars per year.}$$

The fixed charge on the pipe line is

$$\frac{p}{100}\, BLD \text{ dollars per year.}$$

The sum of these two losses is the total yearly loss. Adding and inserting the

values $R = \frac{D}{4}$ and $V = \frac{q}{\frac{\pi D^2}{4}}$ the yearly loss is

$$\frac{16x4Lq^2i}{C^2D^4\pi^2D} - \frac{pBLD}{100}$$

Differentiating this equation with respect to D and setting equal to zero, the minimum value of the expression is

$$\frac{5x6.5xLq^2i}{C^2D^6} - \frac{pBL}{100} = 0$$

from which

$$D = \sqrt[6]{\frac{3250q^2i}{C^2pB}}$$

To determine D we have that q = 635, i = 1750, C = 140, p = 15, and B = 2.

$$D = \sqrt[6]{\frac{3250x(635)^2x1750}{(140)^2x15x2}} = 12.5 \text{ ft.}$$

We shall therefore use a twelve foot pipe with an area of 113 sq. ft., and a full load velocity of 900/113 = 8 feet per sec., which is sufficiently low, for this type of pipe which is to be made of wood staves, with iron bands reinforcing it. The design

of the iron reinforcing bands is shown as follows:

Let A_s = area of rods,/ in. length of pipe.

f_s = allowable stress = 16000#/ sq.in.

p = maximum unit internal pressure = 21.7 lbs., that due to a 150 ft. head.

Therefore

$$2A_sf_s = Dx12xp$$

from which

$$A_s = \frac{12x12x21.7}{2x16000} = 0.097 \text{ sq. in. steel.}$$

The spacing of the bands is , when using a 1 inch band

$$\frac{0.785}{0.097} = 8 \text{ inches.}$$

PENSTOCK.

Because of generating requirements, three units are needed, and each unit will be supplied with it's own penstock. The section area of each penstock is made one-third that of the pipe line, and thus the diameter of the penstocks will be 7 ft.,

each. Since the surge tank is located at the junction of the pipe line and penstocks (connection being made by a reinforced concrete distributor as shown in drawing) all surges caused by gate closing and load fluctuations will travel the entire length of the penstock before being acted upon by the surge tank. Hence the penstocks must be designed for the maximum probable water hammer.

The formula which shall be used here for the water hammer is developed by Mr. M. W. Warren, in the Proceedings of the A. C. E. S., for October, 1914. This formula is more simple and more accurate for moderate penstock velocities than the older formula due to Allievi. The derivation and application of the above mentioned formula is treated in the following paragraphs.

Nomenclature.

T = time of gate closing in seconds.

P = section area of pipe in sq. ft.

V = velocity corresponding to Q in ft./sec.

W = weight per cubic foot of water.

L = length of penstock in feet.

h = head due to water hammer.

a = velocity of vibration along penstock.

Due to variations in time of gate closing some assumption must be made as to the rate of closing. It is assumed that the gate is closed in such a way that in the absence of reflected waves, the pressure will rise at a constant rate. According to this assumption the pressure will rise linearly from 0 to h in a time $\frac{2L}{a}$, remain constant at h until a time T, when it will fall again.

Therefore the impulse(force)(time) acting on the mass of water from time 0 to time T is

$$- PW\left(\frac{h}{2}\right)\left(\frac{2L}{a}\right) - PWh\left(T - \frac{2L}{a}\right) = -PWh\left(T - \frac{L}{a}\right)$$

The mass of water in the pipe is $\frac{PWL}{g}$ and it's velocity changes from V to 0. Therefor the gain of momentum is

$$-\frac{PWLV}{g}$$

Equating the impulse to the momentum we have

$$-PWh\left(T-\frac{L}{a}\right)=-\frac{PWLV}{g}$$

therefore

$$h=\frac{LV}{g\left(T-\frac{L}{a}\right)}$$

To apply the formula just derived we have the following data:

L = 700 ft.

D = pipe diameter = 7 ft.

V = 8 ft. per sec.

T = 2 seconds.

b = pipe thickness in feet, assumed at 0.5"

a = 2830 as calculated below.

$$a=\frac{4660}{\sqrt{1-.01\frac{D}{b}}}=2830 \text{ ft. per sec.}$$

Therefore

$$\frac{L}{a} = \frac{700}{2830} = 0.25 \text{ sec.}$$

$$h = \frac{700 \times 8}{32.2(2.0-.25)} = 100 \text{ ft.}$$

Then the total static head plus the water hammer head is

$$115 + 100 = 215 \text{ ft.}$$

Therefore

$$a_s = \frac{215 \times .434 \times 7 \times 12}{2 \times 16000} = 0.23 \text{ inches.}$$

Therefore a pipe with a thickness of metal of one-half inch will be used for the penstocks.

SURGE TANK.

To provide against destructive surges upon the sudden application or rejection of load some form of stand-pipe or surge-tank is necessary. Previous to the last year or two a large open tank was generally used, and connected so that there was an unobstructed flow between it and the penstock.

This form of regulation while theoretically sound is generally a practical impossibility, due to the excessively large tank required.

To overcome this difficulty Mr. R. D. Johnson has developed a differential tank which not only produces better regulation but is also less expensive to build than the simple tank. A short resumé of the inventors claims which have been proven in at least three recent installations are:

(1) A much smaller diameter than the simple tank.

(2) It does not have to be carried to such a height as the old style tank, thus saving an expensive spillway and waste of water.

(3) It is proof against synchronous load changes, which however small, are likely to carry the water in the simple tank beyond bounds. This last is the most important of all.

We might state here before proceeding further the various conditions which will set a wave motion to vibrating in the penstock. An undulation in the water may be set into motion by a load change, and may have any one of the following four characterstics, depending upon existing conditions.

(1) It may die out as a damped harmonic.

(2) It may live indefinitely at the same amplitude.

(3) It may continue to drop, theoretically, without turning around to rise again, if the acceleration in the pipe is not fast enough to catch up to the increasing draft velocity as the head recedes and the governor opens the water wheel.

(4) It may increase in amplitude with each oscillation.

As a modification of case (1) it may die out with no oscillation at the end of the first quarter cycle in an infinite time. This is called the dead-beat condition.

When load changes happen to occur in step with the natural period of vibration of the water column, case (1) usually takes on the characteristics of (3) and experience has shown that this test applied to simple tanks disqualifies them for safe use. It is known that a differential tank will almost invariably stand up under this test. The variation in case (1) called the dead-beat condition is practically out of the question unless the pipe line be very long or the velocity very high, or a combination of both, either of which is conducive to a large friction factor.

Therefore, a differential tank is chosen for this development, because of its reliability and small initial cost

The formulas as developed by Mr. Johnson and which have been used in this design were published in the October, 1914, issue of the Proceedings of the American

Society of Civil Engineers. The development of the formula will not be taken up here, as it is rather lengthy.

Nomenclature.

t_a = time of acceleration.

t_r = time of retardation.

T = time to complete 1/4 of the wave.

L = length in feet of pipe.

A = area in square feet of the pipe.

F = net pipe area(in excess of the stand-pipe area).

a = area of the restricted opening with a 100% coefficient of discharge.

a_0= area of restricted opening when time is equal to zero.

a_1 = area of restricted opening when time is equal to T.

v = velocity in the pipe.

v_1 = initial pipe velocity before acceleration begins

v_t = pipe velocity when t_a = T = velocity

before retardation begins.

c = coefficient so that cv^2 = total loss in head in the pipe.

y = departure of the water level in the tank from it's intial quiescient position previous to a change in load.

y_1 = y for a time = T = amount of sudden initial change in stand-pipe level.

p = % of velocity change = $\frac{v_2 - v_1}{v_2}$

k = stability factor of wave = $\frac{y}{c(v_2^2 - v_1^2)}$

Z = $\sqrt{\frac{y_1}{c} - v_1^2}$

X = $\frac{Z}{v_2}$

The formulas are given as follows:

$$(1) \quad F = \frac{AL}{2gcvy_1}\left(\frac{v_2}{Z}\log \frac{(Z-v_1)(Z-v_2)}{(Z-v_1)(Z-v_2)} - \log \frac{(Z_2^2 - v_1^2)}{(Z^2 - v_1^2)} \right.$$

$$(2) \quad F = \frac{AL}{2gc^2v_2^2K_r}\left(\log \frac{K_r}{K_r - 1} \right)$$

(3) $a_0 = \frac{A(v_2-v_1)}{\sqrt{2gy_1}}$

(4) $a_1 = \left(\frac{AFy_1}{L}\left(1- \frac{1}{K} \right)\right)^{1/2}$

(5) $v_c = \frac{0.0993}{c}\sqrt{\frac{AL}{F}}$

(6) $K = \frac{y_1}{cv_2^2}$

The computations for the surge tank may be divided into two main divisions.

(a) Dimensions of tank necessary to regulate for a sudden application of load, (from one-half to full load). In the calculations for this condition a certain permissible drop in riser level is assumed, as (y_1 = 10'), and the diameter of the tank and the opening area is calculated for this drop.

(b) Dimensions of tank necessary for a sudden rejection of load, (from full load to no load). With the diameter calculated in part (a), the height of the tank above full load gradient is calculated to take care of

the rise in water due to a rejection of the load. The port area is determined for this condition also, and a compromise is made between the two areas calculated. To determine the dimensions just spoken of the maximum pipe velocity must be known. This is not that due to full load but a critical velocity V_c which may exceed the full load velocity.

The data for the calculation of the tank is given below.

Length of pipe line = 6445 ft.

Diameter = 12 ft.

Area = 113 sq. ft. = riser area.

Full load flow = 900 cusecs.

Half load flow = 450 cusecs.

C = 140

y_1 = 10' (assumed).

For and acceleration from one-half to full load with a drop of ten feet in the riser we have

$$Z = \sqrt{\frac{10}{.11} + 16} = 10.7$$

$$F = \frac{113 \times 6445}{64.4 \times 0.11 \times 10} \left(\log \frac{(10.7-4)(19.7+8)}{(19.7+4)(10.7-8)} - \log \frac{(10.7-16)}{(10.7-64)} \right.$$

$$F = 492.4 \text{ sq. ft.}$$

Therefore the total area is equal to the area of the tank plus the area of the riser, or

$$\text{area} = 492.4 - 113.00 = 605.4 \text{ sq. ft.}$$

Therefore

$$D = \left(\frac{605.4 \times 4}{3.1416} \right)^{1/2} = 27.8 \text{ ft.}$$

This then gives the diameter of a tank necessary to regulate for a drop of 10 ft.

Part (b).

$$a_0 = \frac{113 \times 4}{\sqrt{64.4 \times 10}} = 17.7 \text{ sq. ft.}$$

$$V_c = \frac{0.0993}{0.11}\sqrt{\frac{113x6445}{492.4}} = 34 \text{ ft./ sec.}$$

It can be seen thatthis value is too high, hence a value of ten feet per second, which is two feet per second more than the full load velocity, will be assumed.

From equation (2)

$$F = 492.4 = \frac{113x6445}{64.4x.0121x100K_r} \log \frac{K_r}{K_r-1}$$

By a trial value of K_r the value of 3.15 satisfies the above equation. Substituting in equation (6)

$$y_1 = 3.15x.11x100 = 34.6'$$

which is the height of the tank above the full load hydraulic gradient. From equation (4) we have

$$a = \left(\frac{4.92x113x34.6}{6445} x\left(1 - \frac{1}{3.15}\right) \right)^{1/2} =$$

$$= 14.3$$

Therefore the theoretical area of the opening is 14.5 sq. ft. But the coefficient

of discharge for a rectangular opening is 0.6 (Hughes and Safford Pp. 149) and therefore the above opening must be corrected.

$$\text{area} = \frac{14.5}{0.6} = 24.2 \text{ sq. ft.}$$

The height of the tank is determined by the dimensions just calculated and by the hydraulic gradient at full and half loads.

$$h_f = \text{friction head} = \frac{4V^2L}{C^2D}$$

at half load

$$h_f = \frac{4x16x6445}{19600x12} = 1.75'$$

and at full load

$$h_f = \frac{4x100x6445}{19600x12} = 11' - 00''$$

The forebay elevation being 515' gives a hydraulic gradient at 1/2 load of 513.25' and a hydraulic gradient at full load of 504.00'. The elevation of the tank bottom is

513.25 - 10.00 = 503.25', and the elevation of the top of the tank is 504.00 + 34.60 = 538.60', which gives the height of tank as 538.60 - 503.25 = 35.35 ft.

The calculations and theory for the design of the dam, spillway, pipe line, surge tank and penstocks have now been fully discussed. In the next article the design of the turbines, draft tubes, power house and equipment will be presented.

PART 2.

DESIGN OF POWER HOUSE.

H. A. KLEINMAN.

NOTATION.

A = acceleration.

a = area.

b = constant.

C = constant.

D = diameter in inches.

E = elevation in feet.

e = efficiency.

f = frequency in cycles per second.

g = acceleration due to gravity.

h = head in feet.

K = discharge coefficient.

= constant.

= loss in draft tube in feet head.

l = length of draft tube in feet.

n = speed, revolutions per minute.

P = power, horse power.

= pressure.

p = number of poles.

= power corfficient.

= constant in equation for parabola.

q = discharge in cubic feet per second.

R = radius.

r = radius.

S = distance in feet.

T = time in seconds.

V = velocity in feet per second.

v = velocity in feet per second.

v_1 = velocity at entrance to draft tube.

v_n = velocity at exit of draft tube.

w = weight of one cubic foot of water.

X = distance in feet.

x = coordinates.

y = coordinates.

Z = distances along draft tube.

Z_1 = abscissa at beginning of draft tube.

Z_n = abscissa at end of draft tube.

α = angle.

Δ = speed coefficient.

ϕ = ratio of peripheral velocity to spouting velocity.

γ = specific power.

DESIGN OF POWER HOUSE.

The pipe line, surge tank, and penstocks have been designed to deliver to the power house a maximum of 900 sec. ft., of water under a net head of 180 ft. This hydraulic power will be converted into electrical energy and delivered to a transmission line at 60,000 volts pressure, at 25 cycles. The machinery in the power house will be designed and selected to accomplish this end. To deliver alternating current at 25 cycles at once places restrictions upon the speeds at which the generators may be run, as is shown by the following formula;

$$(1) \qquad f = \frac{np}{2(60)}$$

from which it follows that for f = 25,

$$(2) \qquad n = \frac{3000}{p}$$

As p must necessarily be an even integer

it follows that the generators must be run at one of the following speeds as shown by increasing p by increments of 2 beginning with p = 2: 1500 r. p. m., 750 r. p. m., 500 r. p. m., 375 r. p. m., 300 r. p. m., 250 r. p. m., etc.

For a low head utilizing very large units it may be necessary to run at a low speed thereby increasing the number of poles required on the generators, but with a medium head one of the above speeds would be chosen. After an investigation of operating speeds of various plants utilizing heads of from 125 to 250 ft., and also at the advice of a prominent hydraulic engineer it was decided that 375 r. p. m., was the proper speed for the units in this plant. It is advantageous to use as high a speed as is practicable because the size of the machinery, and it's cost and space requirments are thereby reduced.

The number of units to be installed in any plant depends upon the available power to be utilized and upon the form of the load curve. It is best to use as large units as possible, because the larger units are usually the more efficient; however it would be unwise to install only one large unit, because if at any time repairs were necessary the entire plant would be shut down. Therefore at least two units should be installed, and one of these should be able to carry the load. This would mean that one-half the capacity of the plant would be in the spare unit, thus making the cost of the plant per kilo-watt capacity very high. In this plant a compromise will be effected between the efficiency of the large unit and the reliability of a number of smaller units. Therefore three units will be installed capable of developing about the average

available power of the stream. At low water periods and during periods of normal flow two units only will be run and the third will be a spare unit; during high water it will be possible to run all three units and thus utilize the entire capacity of the plant. In case of accident to one of the units during this time it will of course be necessary to carry the load on the remaining two, but even then the plant will be operating under normal conditions and no serious consequences should arise.

The power available at the power house site may be expressed as:

$$(3) \qquad P = \frac{qwh}{550} = 18400 \text{ H. P.}$$

Assuming a turbine efficiency of about 80%, it can be seen that the turbines should deliver about 15,000 H. P., to the generators. This means that the maximum capacity of each of the three units should be about 5000 H.P. when discharging 300 sec. ft., under a net

head of 180 ft.

The turbines should be considered to be fully loaded when operating at from 0.7 to 0.8 full gate opening; greater gate openings than these are in the nature of overloads. Therefore the normal output of each unit will be about 3750 H.P., and they will be direct connected to the generators rated at 3000 K. V. A., but capable of delivering 25% overload for a short time.

TURBINE ANALYSIS.

The problem of selecting the turbine therefore resolves itself into finding a wheel which, when operating at 0.7 to 0.8 gate, will develope 3750 to 4000 H.P., at its point of maximum efficiency when operating under a head of 180 ft., and running at a speed of 375 r. p. m.

It is of utmost importance for the satisfactory operation of a plant that the proper turbine be installed; therefore, a

detailed discussion of the method of analyzing the performance of a wheel will be given along with the derivations of the relations that exist for wheels of homologous design. Relative to this, D. W. Mead in his WATER POWER ENGINEERING explains it in the following way: " If --- a series of wheels constructed on the same general design with all parts in proportion, so far as practicable, they are termed wheels of ' homologous design ', and constitute a series of wheels which, if really homologous in both design and workmanship, will have common characteristics and hence common relations between their diameter D, and their discharge q, power P, speed n, and efficiency e, under any given effectual head h. These important variables are somewhat independent but have certain interrelations which can be determined by experiment from any wheel of the series, and are expressed by coefficients

from which their corresponding values for a wheel of the series of any given diameter can be determined for any given head and speed ----."

The velocity of the periphery of a turbine may be considered as a function of the spouting velocity of the water, in which case the relation may be expressed as

(4) $$V' = \phi\sqrt{2gh}$$

from which it follows that

(5) $$\phi = \frac{V'}{\sqrt{2gh}} = \frac{V'}{V}$$

Thus ϕ may range in value from zero when the runner is blocked to a maximum when the turbine is running free, but there is only one speed for which the turbine is designed and this is determined by the value of ϕ at the point of maximum efficiency. The efficiency is, of course, zero both when the runner is blocked and when it is running free, it's maximum value being at some speed between these two,

The peripheral velocity may also be expressed in terms of the diameter and the speed; thus

$$(6) \qquad V' = \frac{D\pi n}{12 \times 60} = \frac{3.1416\ Dn}{720}$$

Combining (5) and (6),

$$(7) \qquad \phi = \frac{3.1416\ Dn}{720 \times 8.025\sqrt{h}} = 0.0005437\frac{Dn}{\sqrt{h}}$$

and also

$$(8) \qquad n = \frac{\phi\sqrt{h}}{0.0005437\ D} = \frac{1840\ \phi\sqrt{h}}{D}$$

As equation (8) is general, a speed coefficient may be deduced as follows:

$$(9) \qquad \frac{Dn}{\sqrt{h}} = 1840\ \phi = \Delta$$

If the head is one foot (h=1) it follows that

$$(10) \qquad Dn_1 = 1840\ \phi = \Delta$$

As equation (10) is general it follows that for any fixed value of ϕ there is a

corresponding value of Δ and at these values the diameter of any wheel of a series multiplied by it's corresponding revolutions per minute at any fixed head is constant.

If in equation (10) the diameter is one inch (D=1) the equation becomes

(11) $n_1 = 1840\,\phi = \Delta$

Therefore Δ is equal to the number of revolutions per minute of a one inch wheel under a one foot head.

In any homologous system of turbines the diameters, heights, and corresponding openings and passages being proportional, it follows that similar discharge areas under different gate conditions are proportional to each other and to the squares of any lineal dimension. In such wheels therefore the area $\underline{a}$, of the gate openings is proportional to the square of the diameter of the wheel and the equation may be written

(12) $Ca\sqrt{2g} = KD^2$

In this equation K is a discharge coefficient

which may be determined by experiment. The discharge is given by the formula

$$q = Ca\sqrt{2gh} \tag{13}$$

Combining (12) and (13) it follows that

$$q = KD^2\sqrt{h} \tag{14}$$

If the head equals one foot and the diameter equals one inch, then

$$q = K \tag{15}$$

Therefore K is the discharge of a one inch wheel under a one foot head.

The power of a turbine is

$$P = \frac{qwhe}{550} = \frac{qhe}{8.8} \tag{16}$$

Combining (13) and (16), it follows that

$$P = \frac{Ca\sqrt{2g}\,h^{3/2}e}{8.8} \tag{17}$$

From equation (17) it is apparant that if C, e, and a are constant for any given turbine and fixed gate opening and if the value of ϕ remains the same, the power of the turbine will be in direct proportion

to $h^{3/2}$. By substituting equation (14) in equation (16) it follows that

$$(18)\quad P = \frac{D^2h^{3/2}Ke}{8.8} = \frac{Ke}{8.8}D^2h^{3/2}$$

As $\frac{Ke}{8.8}$ is constant for a given wheel as long as ϕ is constant this expression may be represented by a constant p which is termed the " power coefficient". With this substitution,

$$(19)\quad P = pD^2h^{3/2}$$

If the diameter equals one inch (D=1) and the head is one foot (h=1), then

$$(20)\quad P=p$$

Therefore p is the power of a wheel one inch in diameter operating under a one foot head.

There are certain relations of speed and power which are important in the preliminary selection of a turbine for a given work. From equation (9)

$$(21)\quad D = \frac{\Delta\sqrt{h}}{n}$$

and from equation (19)

$$(22)\quad D^2 = \frac{P}{ph^{3/2}}$$

By equating the values of D^2 it follows that

$$(23)\quad \frac{\Delta^2 h}{n^2} = \frac{P}{ph^{3/2}}$$

and therefore

$$(24)\quad \Delta^2 p = \frac{n^2 P}{h^{5/2}}$$

The combined coefficient $\Delta^2 p$ is represented by the symbol $\wp$ which may be termed " specific power".

$$(25)\quad \wp = \Delta^2 p = \frac{n^2 P}{h^{5/2}}$$

and when the head is one foot (h=1) and the diameter is one inch (D=1) then

$$(26)\quad \wp = P$$

which means that $\wp$ is the power of a wheel of such a diameter that it will make one revolution per minute under one foot head.

To summarize, the following coefficients

have now been established:

1.,from equation (9)

$$\Delta = \frac{Dn}{\sqrt{h}}$$

where Δ is termed the speed coefficient and is equal to the number of revolutions per minute of a one inch wheel under a one foot head.

2.,from equation (14)

$$K = \frac{q}{D^2\sqrt{h}}$$

in which K is the discharge coefficient and is equal to the discharge of a one inch wheel under a one foot head.

3.,from equation (19)

$$p = \frac{P}{D^2\, h^{3/2}}$$

where p is the power coefficient and is equal to the power of a wheel one inch in diameter operating under a one foot head.

4.,from equation (25)

$$\mathfrak{g} = \frac{n^2P}{h^{5/2}}$$

in which $\mathfrak{g}$ is the specific power and is equal to the power of a wheel of such diameter that it will make one revolution per minute under one foot head.

In order to make use of these coefficients in selecting a wheel to fulfill any requirements under a given set of conditions, it is necessary to have the results of tests of wheels of various types, that is, of wheels designed to run at maximum efficiency at different values of ϕ in equation (5). The conditions under which a wheel is designed to operate determines the value of ϕ; for example, a wheel designed to run at a comparatively low speed under medium or high heads would require that the ratio of the peripheral velocity of the runner to the spouting velocity of the water be small, (thus that ϕ be small); conversely, a wheel designed to run at a high speed under

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

low or medium heads would have a high value of ϕ.

In Appendix B, of Mead's "Water Power Engineering" are given the results of a number of tests run at the Holyoke testing flume on wheels of various types and sizes. From these test sheets the specific power is calculated for each run from equation (25)

$$\mathfrak{z} = \frac{n^2P}{h^{5/2}}$$

A curve is then plotted for each wheel showing the relation between specific power and efficiency. From the conditions of this problem it can be shown that the specific power should equal

$$\mathfrak{z} = \frac{(375)^2 \text{ x } (3750)}{(180)^{5/2}} = 1215$$

for rated loading and

$$\mathfrak{z} = \frac{(375)^2 \text{ x } (5000)}{(180)^{5/2}} = 1620$$

for 25% overload. Therefore from the specific

power curves for the various wheels it is necessary to pick out a wheel which has it's point of maximum efficiency at a value of specific power equal to about 1250. Upon investigation of the wheels given in Appendix B there were found to be no wheels which would prove entirely satisfactory for this plant. A letter was addressed to the Wellman-Seaver- Morgan Co., stating the conditions for which a wheel was desired and the reply from the Chief hydraulic engineer of this company recommened the use of a runner manufactured by this company and designated as "Runner # 11". A test data sheet showing the results of tests made on a runner of this type at the Holyoke testing flume was enclosed and an investigation of this wheel showed that it would be satisfactory.

In Table 1, on the next three pages, are shown the results of the tests on this wheel and also the values of Δ and p calcu-

Table 1.

TURBINE ANALYSIS

TESTING FLUME OF THE HOLYOKE WATER POWER CO.

TEST OF A 32" L.H. WELLMAN-SEAVER-MORGAN CO. TURBINE WHEEL

NO. OF RUN	GATE OPENING	NET HEAD ACTING ON WHEEL	SPEED	POWER DEVELOPED BY THE WHEEL	EFFICIENCY OF THE WHEEL	SPEED COEFFICIENT Δ	POWER COEFFICIENT p
	PERCENT	FEET	R.P.M.	H.P.	PER CENT	$\Delta = \frac{Dn}{\sqrt{h}}$	$p = \frac{P}{D^2 h^{\frac{3}{2}}}$
75	100.0	16.34	147.33	85.11	80.32	1168	.001255
74	100.0	16.39	155.00	84.83	80.63	1228	.001248
73	100.0	16.38	164 33	84.95	81.50	1300	.001249
72	100.0	16.37	173.25	84.28	81.53	1370	.001242
71	100.0	16.35	183.25	83.58	81.80	1450	.001233
70	100.0	16.35	193.50	82.36	81.19	1530	.001215
69	100.0	16.36	204 00	80.63	79.64	1615	.001190
68	100.0	16.33	214.00	78.08	78.01	1695	.001152
67	100.0	16.25	225.00	72.51	73.99	1780	.001080
66	100.0	16.55	296.75	—	—	2350	—
58	86.6	15.82	140.25	75.05	81.70	1128	.001165
57	86.6	15.53	145.50	73.43	82.74	1180	.001170
56	86.6	15.61	150.00	72.97	82.40	1218	.001152
59	86.6	16.16	174.25	79.47	85.97	1380	.001193
62	86.6	16.13	180.50	79.57	86.57	1430	.001200
60	86.6	16.18	187 20	79.68	86.57	1480	.001196
61	86.6	16.16	199.25	78 75	86.24	1578	.001185
63	86.6	16.14	208.67	76.13	84.35	1647	.001145
64	86.6	16.88	225 50	72.67	77.13	1780	.001022
65	86.6	16 89	299.75	—	—	2360	—
50	75.6	16.39	159.25	72.63	85.65	1258	.001068
51	75.6	16.40	169.75	74.32	87.69	1340	.001093
52	75.6	16.39	175.25	74.60	88.14	1385	.001098
53	75.6	16.38	181.75	75.15	89.13	1430	.001105
54	75.6	16.36	187.00	75.05	89.37	1478	.001104
55	75.6	16.32	191.25	73.27	88.28	1510	.001075
40	73.6	16.62	136.00	70.29	81.27	1042	.001010
39	73.6	16.64	145.75	71.79	83.55	1118	.001032

Table 1

TURBINE ANALYSIS (2)

NO. OF RUN	GATE OPENING	NET HEAD ACTING ON WHEEL	SPEED	POWER DEVELOPED BY THE WHEEL	EFFICIENCY OF THE WHEEL	SPEED COEFFICIENT Δ	POWER COEFFICIENT p
	PER CENT	FEET	R.P.M	H.P.	PER CENT	$\Delta=\frac{Dn}{\sqrt{h}}$	$p=\frac{P}{D^2h^{\frac{3}{2}}}$
38	73.6	16.65	153.50	72.80	85.28	1179	.001045
37	73.6	16.66	160.25	73.08	86.17	1230	.001050
41	73.6	16.62	171.00	74.87	88.58	1312	.001075
42	73.6	16.62	176.75	75.23	89.28	1350	.001080
43	73.6	16.61	182.00	75.26	89.63	1397	.001080
44	73.6	16.58	187.50	74.11	89.23	1440	.001065
45	73.6	16.57	196.00	71.51	87.48	1503	.001028
46	73.6	16.58	203.33	68.00	84.54	1558	.000978
47	73.6	16.58	211.67	64.36	81.35	1620	.000923
48	73.6	16.63	229.75	55.88	72.73	1760	.000803
49	73.6	16.89	293.50	—	—	2250	—
25	67.0	16.78	131.00	63.73	79.45	1020	.000905
24	67.0	16.78	146.00	66.58	83.97	1138	.000945
27	67.0	16.92	152.50	68.62	85.19	1190	.000962
26	67.0	16.95	157.50	69.43	86.15	1228	.000970
23	67.0	16.78	159.50	67.89	86.72	1243	.000964
31	67.0	16.94	166.75	70.47	87.94	1300	.000983
28	67.0	16.92	172.75	70.38	88.59	1347	.000985
29	67.0	16.91	178.50	69.47	88.35	1390	.000975
30	67.0	16.92	185.75	67.77	87.07	1448	.000948
32	67.0	17.08	197.25	65.97	85.35	1540	.000910
33	67.0	17.12	207.75	63.16	82.80	1612	.000870
34	67.0	17.13	219.25	59.99	80.11	1710	.000827
35	67.0	17.14	228.50	55.58	75.34	1780	.000765
36	67.0	17.33	291.75	—	—	2210	—
15	52.0	17.08	127.50	48.07	77.40	987	.000663
14	52.0	17.07	145.50	51.32	82.35	1125	.000710
13	52.0	17.11	154.25	51.59	83.13	1192	.000713

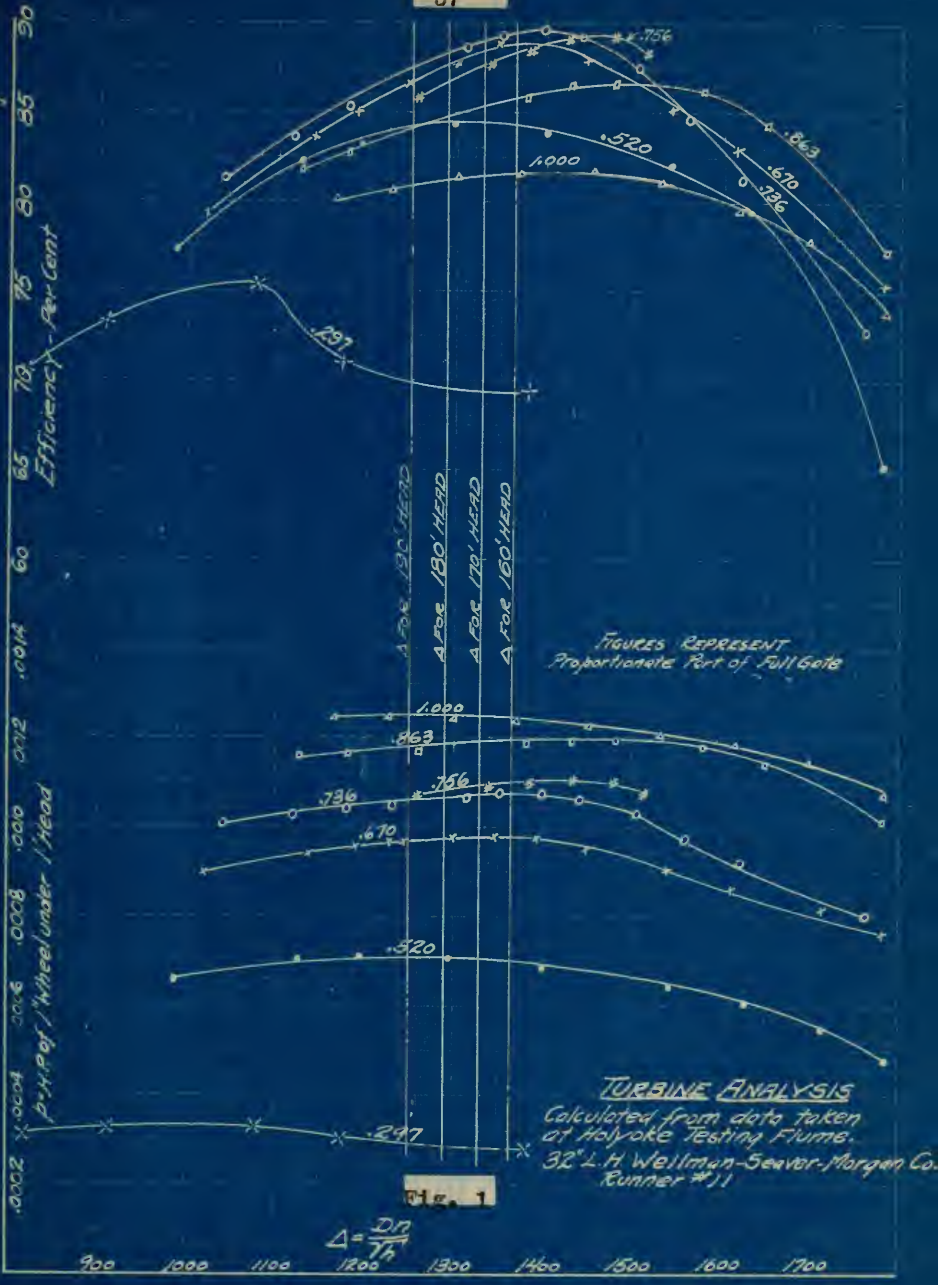
Table 1

TURBINE ANALYSIS (3)

NO. OF RUN	GATE OPENING	NET HEAD ACTING ON WHEEL	SPEED	POWER DEVELOPED BY THE WHEEL	EFFICIENCY OF THE WHEEL	SPEED COEFFICIENT Δ	POWER COEFFICIENT p
	PER CENT	FEET	R.P.M	H.P.	PER CENT	$\Delta = \frac{Dn}{\sqrt{h}}$	$p = \frac{P}{D^2 h^{\frac{3}{2}}}$
22	52.0	17.01	153.25	51.25	83.29	1190	.000712
21	52.0	17.01	167.50	50.93	84.38	1294	.000708
20	52.0	17.03	181.00	49.53	84.05	1400	.000687
16	52.0	17.07	199.25	47.25	82.12	1540	.000650
19	52.0	17.14	210.67	44.84	79.52	1625	.000615
17	52.0	17.12	221.50	40.41	73.16	1710	.000552
18	52.0	17.19	230.67	35.07	65.22	1780	.000480
7	52.0	17.19	272.50	—	—	2100	—
6	29.7	17.73	107.50	24.84	70.47	820	.000324
5	29.7	17.75	120.00	25.54	73.34	914	.000332
4	29.7	17.80	141.00	25.79	75.47	1078	.000335
3	29.7	17.80	154.00	23.41	71.06	1172	.000304
2	29.7	17.84	181.50	22.07	69.67	1380	.000286
1	29.7	17.96	259.00	—	—	1970	—

lated from equations (9) and (19), for each run. The values are plotted as shown in Fig. 1, with values of Δ as abscissa and values of p and e as ordinates. The curves for the various gate openings show that for 0.736 gate opening the maximum efficiency occurs for a value of Δ equal to 1400 and that for 0.670 gate the value is 1365. The operating head may vary from 160 to 190 feet; therefore the characteristics of the wheel will be determined for the following heads: 160 ft., 170 ft., 180 ft., and 190 ft. The head of 160 ft., represants the worst conditions at times of lowest water; therefore it is best to have the highest efficiency at this head. Taking a value of Δ equal to 1365, which gives the maximum value of efficiency for a gate opening of about 0.7, the diameter of the wheel required can be determined from equation (9) by solving for D.

$$D = \frac{(1365)\sqrt{160}}{375} = 46''$$



FIGURES REPRESENT
Proportionate Part of Full Gate

TURBINE ANALYSIS
Calculated from data taken
at Holyoke Testing Flume.
32" L.H. Wellman-Seaver-Morgan Co.
Runner #11

Fig. 1

Therefore a 46" wheel will be used. With a wheel of this size Δ takes the values 1365 for 160 ft., head, 1330 for 170 ft., head, 1290 for 180 ft., head, and 1250 for 190 ft., head. The vertical lines on Fig. 1, give the position of these values of Δ and are labeled to show the head at which the value holds.

From the values of p at the intersection of the power coefficient curves with the vertical lines representing the values of Δ for the four values of head, the power of the wheel is calculated from equation (19). Table 2 shows tabulated results of the power, efficiency, and discharge at various gate openings for the four values of head. The discharge may be calculated from either equation (14) using the discharge coefficient, or knowing the efficiency, it may be calculated from equation (16).

$$P = \frac{qwhe}{550}$$

Table 2.

TURBINE ANALYSIS

PERFORMANCE OF A 46" WHEEL RUNNING AT 375 R.P.M. UNDER HEADS AS NOTED.

1. HEAD = 160 FT. Δ = 1365

GATE-%	p	P	E	Q
100.0	.00125	5360	81.7	360
86.3	.00119	5100	85.8	327
75.6	.00110	4710	88.0	294
73.6	.00108	4620	89.5	284
67.0	.00098	4200	89.0	260
52.0	.00070	3000	84.3	196
29.7	.00028	1200	69.5	95
2. HEAD = 170 FT. Δ = 1330				
100.0	.00125	5840	81.6	370
86.3	.00118	5500	85.5	333
75.6	.00110	5130	87.5	304
73.6	.00107	5000	89.2	290
67.0	.00098	4570	88.7	266
52.0	.00070	3270	84.5	200
29.7	.00028	1310	69.6	97
3. HEAD = 180 FT. Δ = 1290				
100.0	.00125	6370	81.5	382
86.3	.00118	6000	84.7	346
75.6	.00109	5560	86.8	313
73.6	.00107	5450	88.5	300
67.0	.00098	5000	88.0	277
52.0	.00071	3620	84.5	203
29.7	.00029	1480	69.5	104
4. HEAD = 190 FT. Δ = 1250				
100.0	.00125	6930	81.0	396
86.3	.00118	6540	84.0	360
75.6	.00107	5930	85.8	320
73.6	.00105	5820	87.5	307
67.0	.00097	5380	86.8	287
52.0	.00071	3940	84.5	215
29.7	.00030	1665	70.0	110

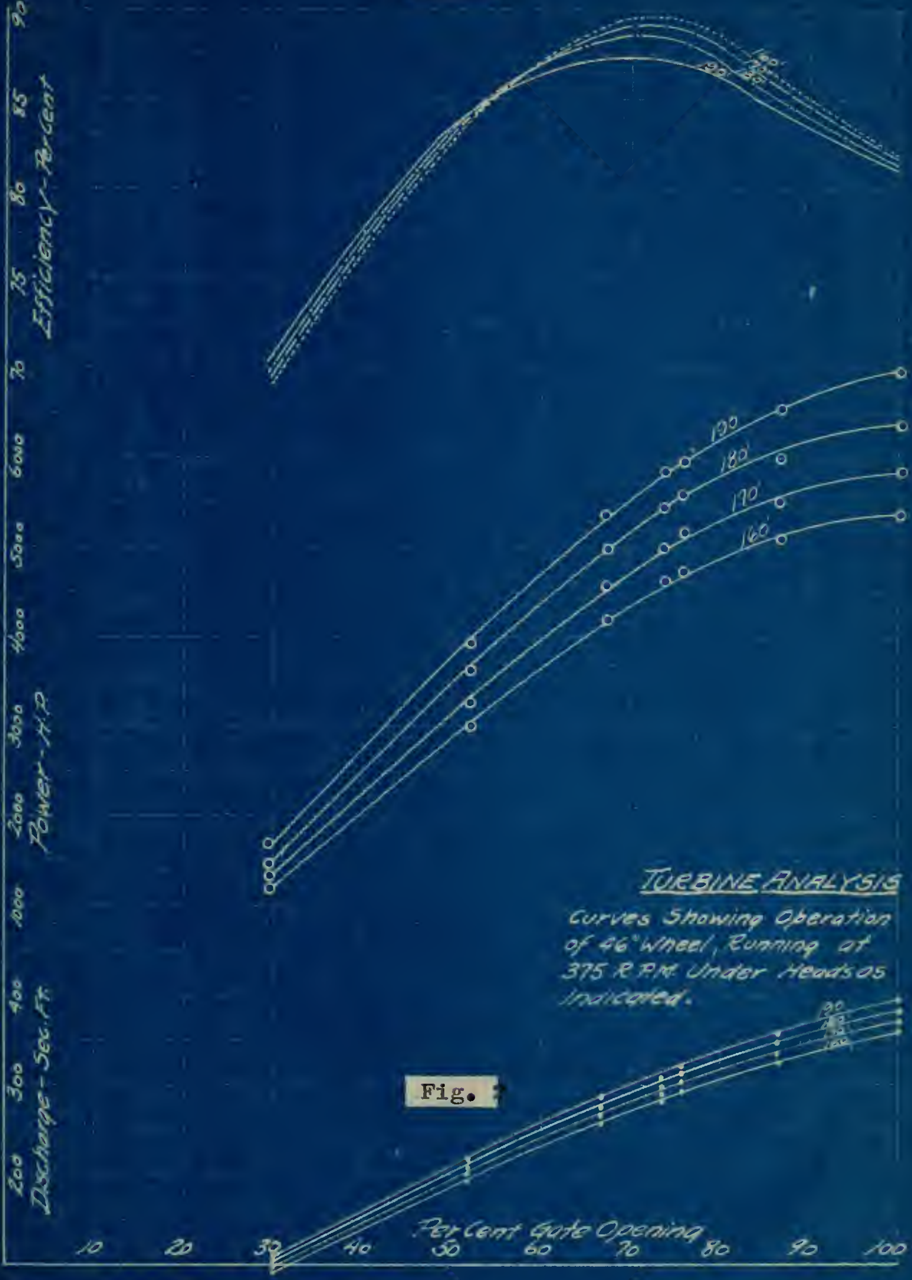
Solving for q,

$$q = \frac{550P}{whe}$$

In this table equation (16) was used in calculating the discharge.

In Fig. 2, the results shown in Table 2 are plotted, showing the efficiency, power, and discharge for any gate opening at any of the four values of head.

In Fig. 3 is shown the relation between efficiency and power, from which it can be seen that this wheel will answer the requirements of this plant in a very satisfactory manner. When running under a head of 160 ft., it will develope from 3750 to 5000 H. P., at an efficiency in excess of 87.5%. When operating under higher heads within this same range of power the efficiency decreases, but as the water is more plentiful and the decrease is not excessive this wheel should prove satisfactory.



TURBINE ANALYSIS

Curves Showing Operation of 46" Wheel, Running at 375 R.P.M. Under Heads as Indicated.

Fig.

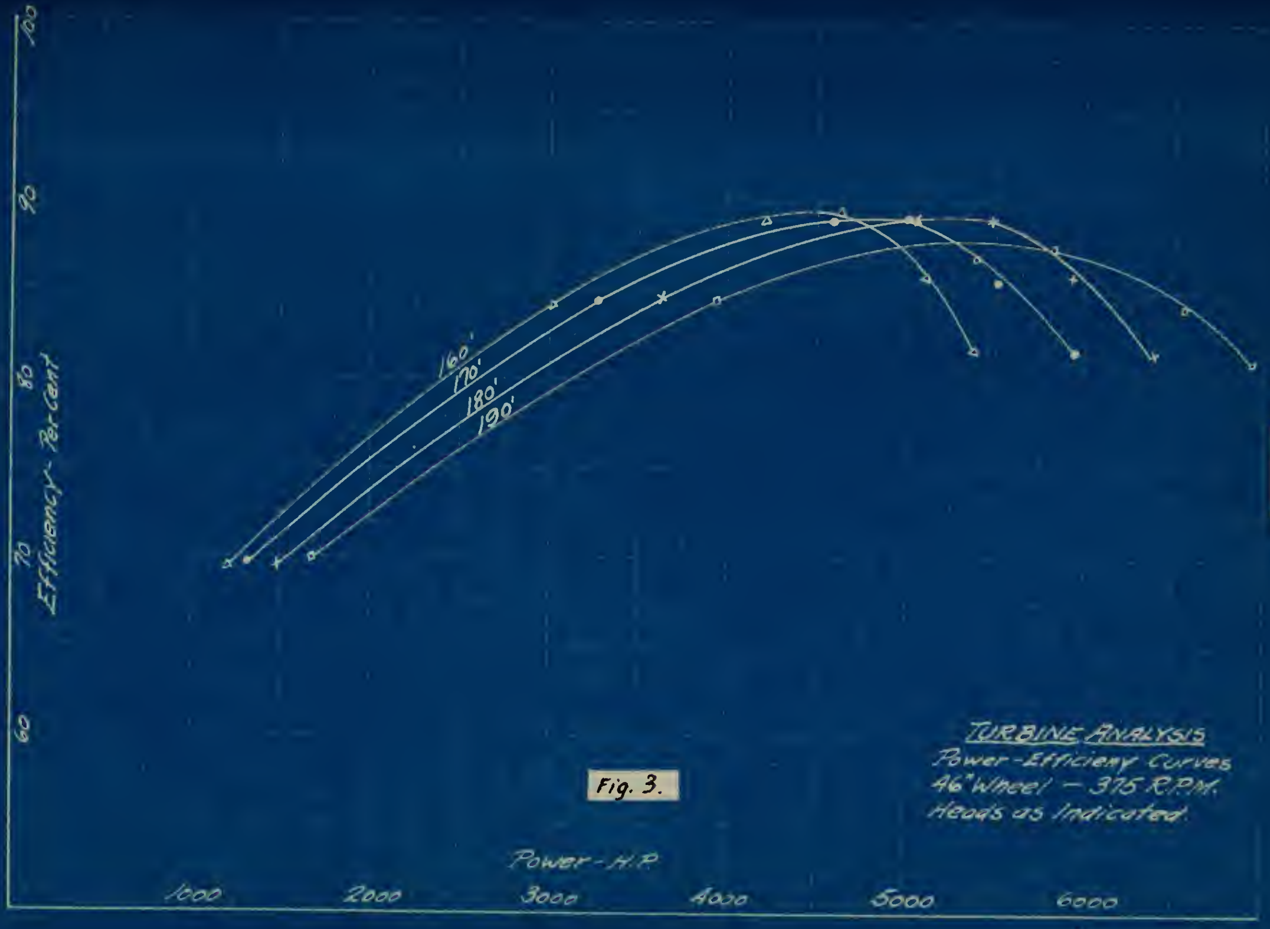


Fig. 3.

TURBINE ANALYSIS
Power-Efficiency Curves
46" Wheel – 375 R.P.M.
Heads as Indicated.
Efficiency - Per Cent
100
90
80
70
60
Power - H.P.
1000
2000
3000
4000
5000
6000
160'
170'
180'
190'

From these curves it has been shown that a wheel homologous in design to the one tested and 46" in diameter will answer the requirements of this plant. It will run at 375 r. p. m., at 0.7 gate, developing about 4200 H. P., under 160 ft., head. It can be seen that it's capacity at the higher heads is larger than required, but as the efficiency is high even at 0.5 gate it will be satisfactory to run at smaller gate openings, and then in case a large amount of power is required for a short period of time it will be available.

DESIGN OF SCROLL CASE.

The efficiency of a hydraulic installation depends upon the design of the intakes scroll case and draft tube almost as much as it does upon the runner itself. For reaction, wheels operating under medium and high heads, the spiral casing is recognized as the best

means available for conducting the water around the wheel. The casing is tapered to keep the pressure constant, in order to feed the water in equally on all sides of the wheel and thus prevent unequal wear on the bearings.

In low- or medium-head single-runner installations, the passage for the water can be divided into five main parts: intakes, scroll case, speed ring, turbine and draft tube. The term 'intake' signifies the passage from the head gates up to the point where the water enters the scroll case. The design of the intakes for this development has been treated elsewhere, and the scroll case proper will now be considered.

Experience has established the best velocity of flow in the scroll case as from 0.15 to 0.20 the spouting velocity of the water, $\sqrt{2gh}$, where h is the effective head of the plant. The velocity of flow in the

scroll case can be determined from the formula

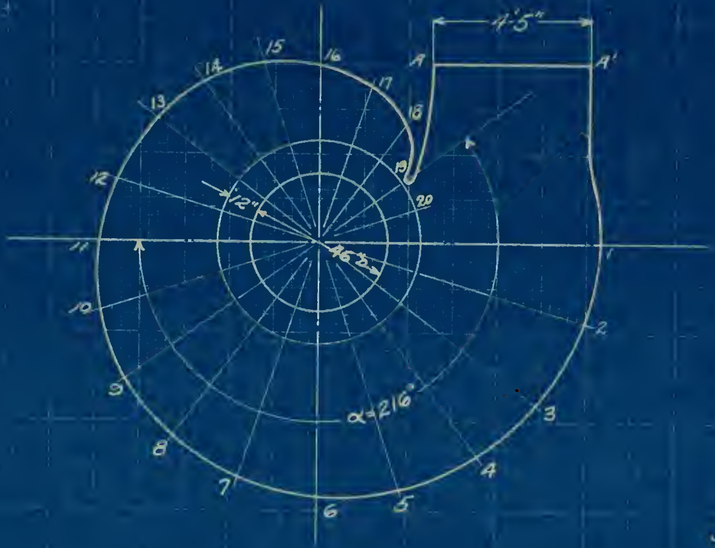
$$v = C\sqrt{2gh}$$

where C = 0.15 to 0.20, and g = 32.16, and h the net operating head. For this development with a net head of 180 ft., a coefficient C = 0.18 was selected . Therefore the velocity in the scroll case is

$$v = 0.18\sqrt{2 \times 32.16 \times 180} = 19.6'/\text{sec.}$$

This velocity will be maintained constant throughout the scroll case.

The point where the scroll case actually begins depends upon the angle α in the Fig. 4. Theoretically the best value for α is 180 degrees, but practically it is better to make α larger and thus economize on space, because there is practically no difference in efficiency for values of α between 180 and 270 degrees. In this design α will be taken as 216 degrees. There being 20 guide vanes, it follows that for a passage of 300



SCROLL CASE
Scale - ¼" = 1'-0"
Discharge - - 300 Sec.-Ft.
Velocity - - - 19.6 Ft. per Sec.

Fig. 4

sec. ft., through the wheel provision must be made for 270 sec. ft., on the transverse axis of the unit at section No. 1, Fig. 4, and for 120 sec. ft., on the same axis but at section No. 11. In Table 3 the discharge at various sections numbered as in Fig. 4 are given, along with dimensions of sections and distance as explained later.

Theoretically, the best shape for cross sections of a scroll case is circular. The hydraulic radius is then a maximum. As these units are comparatively small and as the generators will require a spacing of 20'-00" center to center, there will be room for and it will be advantageous to use circular sections for the scroll case. These sections are made symetrical about the horizontal center line of the distributor.

In passing from the casing proper through the speed ring the water is accelerated to from $0.6\sqrt{2gh}$ to $0.8\sqrt{2gh}$ at which

Table 3.

SCROLL CASE

No. of Point	Q Sec.-Ft.	Area of Section Feet²	Diam. of Section Feet	Distance X Inches	Diam. of Section Inches
A-A'	300	15.40	4.43		53.0
1	270	13.85	4.20	33.0	50.5
3	240	12.30	3.96	31.4	47.5
5	210	10.80	3.70	29.6	44.5
7	180	9.23	3.43	27.6	41.0
9	150	7.70	3.13	25.5	37.5
11	120	6.15	2.80	23.0	33.5
13	90	4.60	2.42	20.2	29.0
15	60	3.08	1.98	16.7	24.0
17	30	1.54	1.40	11.2	17.0
19	0	0.00	0.00	0.0	0.0

Table 4.

TAPER PIECE

No. of Point	Time Seconds	V Feet per Sec.	A Feet per Sec.²	Area of Section Square Feet	Diam. Feet	Distance from Zero Point Feet
1	0	7.8	0.	38.5	7.00	0.
2	0.1	8.2	7.40	36.6	6.83	.79
3	0.2	9.2	13.80	32.6	6.44	1.67
4	0.3	10.9	18.60	27.5	5.92	2.67
5	0.4	12.9	20.90	23.3	5.45	3.85
6	0.5	15.2	20.30	19.7	5.00	5.27
7	0.6	16.7	17.60	18.0	479	6.85
8	0.7	18.7	11.00	16.0	4.52	8.74
9	0.876	19.6	0.	15.4	4.43	12.00

velocity it enters the turbine. The upper and lower rings of the speed ring will have curves conforming to arcs of circles of constant radius and tangent to the section of the scroll case and to the top and bottom of the speed ring as shown in Fig. 5. In order that this condition may be complied with the distance X, Fig. 5, must be made according to the equation

$$X = \sqrt{R^2 + 24R - 151.84}$$

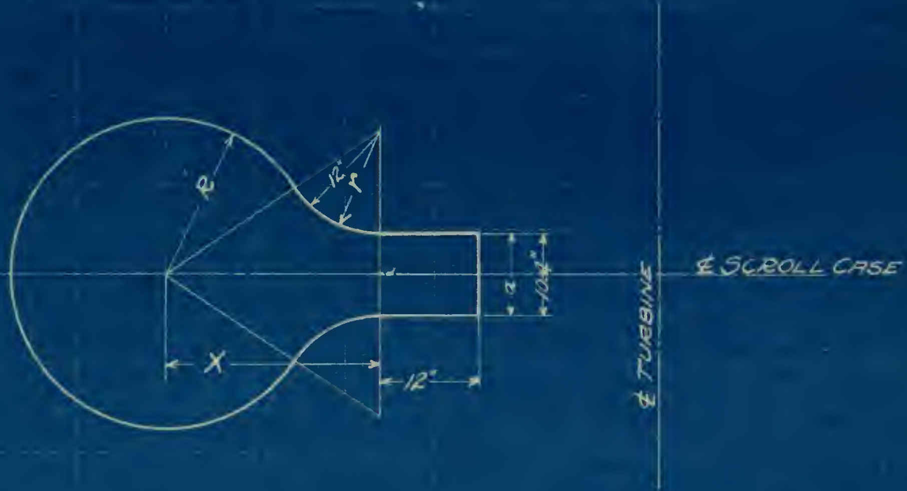
derived for a value of 12" for the radius of the curves for the upper and lower speed speed rings. From the geometry of the figure it can be seen that

$$X^2 + (\frac{a}{2} + r)^2 = (R + r)^2$$

and for values of r = 12 and a = 10.25 it follows that

$$X = \sqrt{R^2 + 24R - 151.84}$$

In this way an efficient bell mouth is created.



TYPICAL SECTION OF
SCROLL CASE

$$X = \sqrt{R^2 + 24R - 151.84}$$

Fig. 5.

For this development the horizontal center line of the scroll case will be located at elevation 345 ft., which is three feet above high water and fifteen feet above low water. It is desirable to have the wheel above high water in order that it will be accessible for repairs without necessitating the construction of coffer dams and the pumping out of the wheel pit. In order to determine whether or not this will be satisfactory it is necessary to investigate certain factors. The determining factor is atmospheric pressure, which will be taken at 33.9 ft. The total draft head must be made less than the theoretic barometric column in order that a margin be allowed:

1. for governing purposes, as a sudden closing of the turbine gates tends to increase the vacuum immediately below the runner and thus a margin must be available in order to prevent the total draft head from

exceeding the height of the barometric column.

2. for the vapor tension of the water, which for ordinary temperature is less than 1 ft., of water.

Because of the losses in the draft tube due to friction and curvature and the outflow losses from the draft tube, all of which tend to decrease the total actual draft head, it is possible to increase the distance of the turbine above the tail water.

It is customary to measure the static head to the top of the runner band. The velcity inside the band of the runner at this point is calculated so that a maximum total draft head is obtained. Applying Bernoulli's Theorem

$$D = E + \left(\frac{V^2}{2g} \right) - L$$

where D = total draft head at the runner;

E = difference in elevation between

the top of the runner band and lowest tail water;

L = loss;

V = velocity;

In this case E = 15; V = 26 ft./ sec., and $M-L = 33.9 - 15 - \frac{26^2}{64.4} = 8.4$ ft., where M is the margin allowed. This quantity M-L is acceptable as it is ussually assumed at from 3 to 6 ft. Therefore, the elevation 345 ft., is satisfactory and will place the wheel above water at all times.

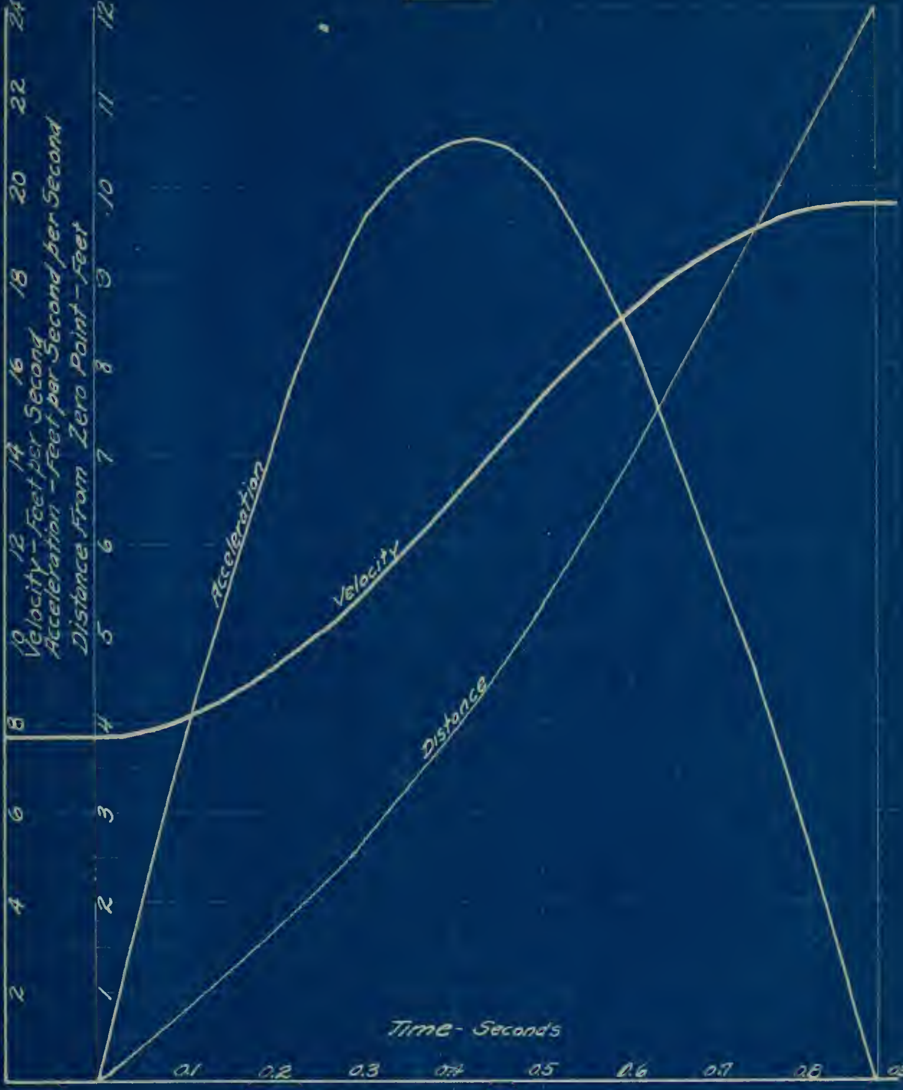
TAPER PIECE.

The penstocks are 7' - 0" in diameter and the entrance to the scroll case is 4'- 5" in diameter. It is necessary, therefore, to connect these with some form of tapered section which will increase the velocity from penstock velocity to scroll case velocity with the least loss of head. The velocity in the penstocks is 7.8 ft., per sec.,

and in the scroll case it is 19.6 ft., per sec., therefore, a change in velocity of 11.8 ft., per sec., is required. It is desired to have the acceleration zero at the beginning of this tapered section, and raise gradually along some curve to a maximum value and then decrease to zero at the end of the section. A sine wave will be assumed as a satisfactory curve for the acceleration. If the length of the tapered section is to be 12'-0", it will take a particle of water 0.876 sec., to pass thru it at an average velocity of

$$V = \frac{7.8+19.6}{2} = \frac{27.4}{2} = 13.7 \text{ ft./sec.}$$

With the acceleration in the form of a sine curve, the velocity curve will be a form of cosine curve and should be such as to increase the velocity gradually from 7.8 to 19.6 ft./sec. Assuming axes as shown in Fig. 6, it can be shown that the equation of such



$$V = 13.7 - 5.9\, \mathrm{Cos}\, \frac{\pi}{.876} T$$
$$A = 21.1\, \mathrm{Sin}\, \frac{\pi}{.876} T$$
$$S = 13.7\, T - 1.64\, \mathrm{Sin}\, \frac{\pi}{.876} T$$

TAPER PIECE
Velocity and Acceleration
Curves

Fig. 6.

a curve may be written as

$$(27) \quad V = 13.7 - 5.9\cos\left(\frac{3.1416}{0.876}T \right)$$

The acceleration curve will be the first derivative of the velocity curve with respect to time;

$$(28) \quad A = \frac{dv}{dt} = 21.1\sin\left(\frac{3.1416}{0.876}T \right)$$

The distance travelled by a particle of water in any time, when following the velocityy curve shown in equation (27) is equal to the integral of this curve between the time limits within which this distance is desired; thus

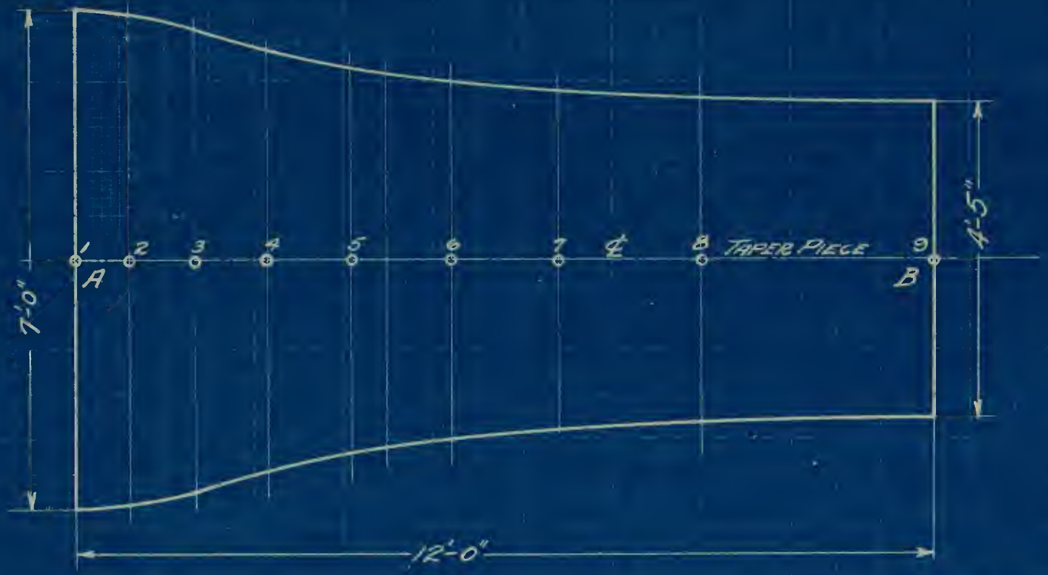
$$(29) \quad S = \int_{T_1}^{T_2} V\, dT$$

or in this case, as the lower limit will always be taken as zero, it is equal to

$$(30) \quad S = \int_{0}^{.876} \left(13.7-5.9\cos\left(\frac{3.1416}{0.876}T\right)\right)dT$$

$$S = 13.7\,T-1.64\sin\left(\frac{3.1416}{0.876}T\right)$$

From these equations for velocity,



TAPER PIECE

Scale-$\frac{1}{2}''$=1'-0"

Discharge ---300 Sec.-Ft.

Velocity at A- 7.8 Ft. per Sec.

Velocity at B - 19.6 Ft. per Sec.

Fig. 7.

acceleration, and distance the curves shown in Fig. 6 are plotted, and Table 4 is calculated, showing the cross-sectional areas, diameters at various sections and the distances from the beginning of the tapered piece at which these sections should be placed. From this table the form of the tapered piece is determined and is shown in Fig. 7.

DRAFT TUBE.

The problem of designing a draft tube does not permit of exact mathematical treatment and some assumptions are necessary in order to get a starting point in the analysis. Therefore it will be assumed that:

(a). the flow in the tube, because of the slight increase in areas from one section to the other is normal to the plane under consideration;

(b.) the change in the velocity head should not at any time be greater than the

change in elevation;

Starting with these assumptions the following factors will be investigated in the order named:

(a.) Shape and length of the center line.

(b.) Velocities and change in velocity heads.

(c.) Shape of areas of cross section.

The velocities, areas, and centers of the beginning and end of the draft tube are known or can be easily determined. In this case the end section will have its center at a point 18 ft., below the intersection of the cent er line of the runner and the center line of the scroll case. The outflow velocity will be 4 ft./sec., and therefore in order to discharge 300 sec. ft., the end section should have an area of 75 sq. ft. The beginning section is on the center line of the runner band. It is, therefore,

necessary to establish some form of center line which will connect these two points. There are several curves which might be used; namely, areas of circles, ellipses, parabolas, or some form of curve of a higher order. The parabola is a very good because it is in accordance with the second assumption, that the change in velocity head should not exceed the change in elevation. Thus, where the change in velocity heads is greatest the center line will be the steepest and where the change in velocity head per unit length becomes small the parabola will come closer and closer to the horizontal. In this case the center line will be composed of three sections; first, a straight section dropping down from the beginning of the draft tube to a point at elevation 339 ft.; at this point the parabolic section will start with the apex of the parabola. In order to bring the center

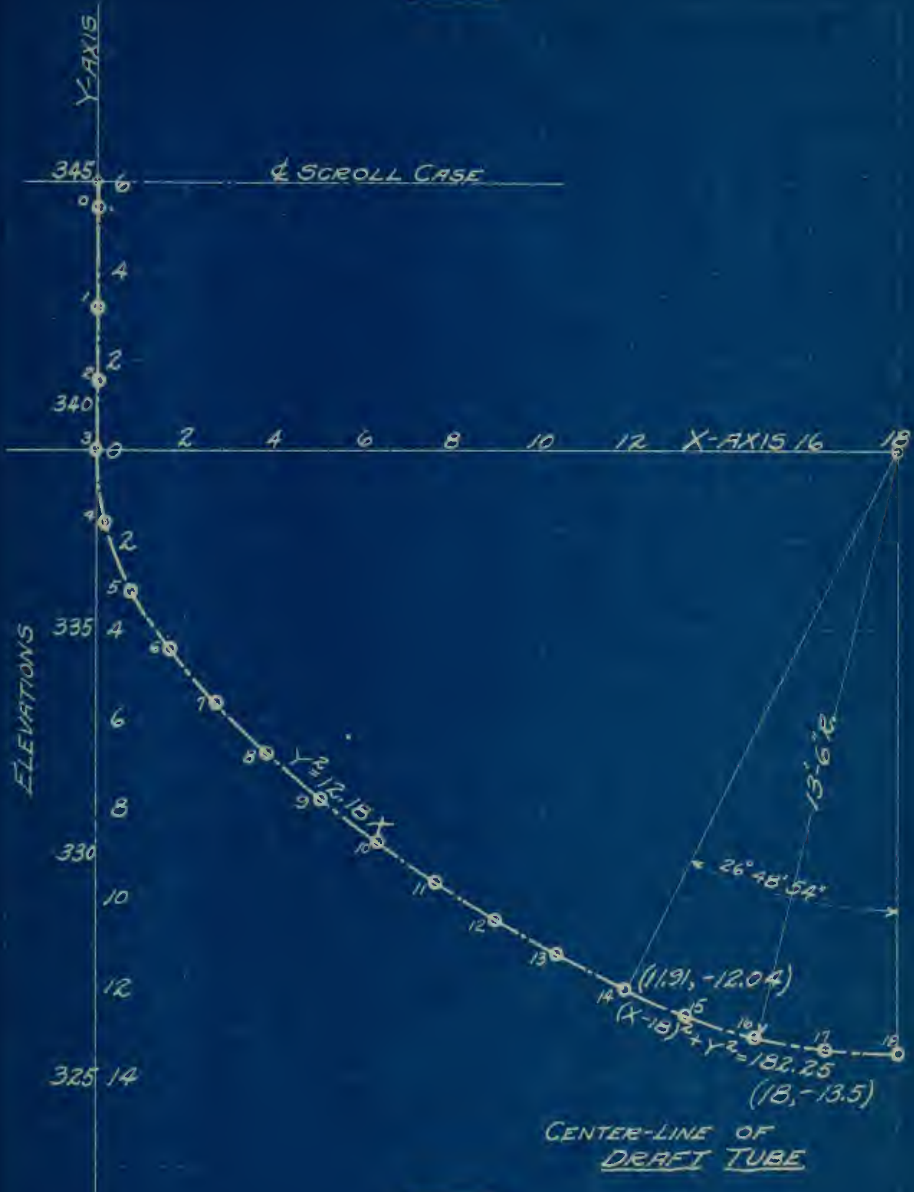

Y-AXIS
℄ SCROLL CASE
ELEVATIONS
0 2 4 6 8 10 12 X-AXIS 16 18
Y² = 12.18 X
(11.91, -12.04)
(X-18)² + Y² = 182.25
(18, -13.5)
26° 48' 54"
13'-6" R.
CENTER-LINE OF DRAFT TUBE

Fig. 8.

line around until it is horizontal at the center of the end section, an arc of a circle will be used as a third section. The form of this center line is shown in Fig. 8. It is necessary to find a parabola such that it will have its apex at (0,0) and will become tangent to some circle which has its center somewhere on the line, $X = 18$. Assuming that the center of the circle will be at point (18,0), the equation of this circle, remembering that it must pass through point (18,13.5), is

(31) $(X-18)^2+Y^2 = 182.25$

It is now necessary to find some parabola which will have its apex at the origin and will be tangent to the circle just determined. The general equation of a parabola with apex at the origin and lying along the X axis is

(32) $Y^2 = 2pX$

Solve equations (31) and (32) simultaneously to find points of intersection, thus:

substututing (32) in (31)

$$(X-18)^2+2pX = 18225$$

solving for X

$$X^2-36X+324+2pX = 182.25$$

$$X^2+(2p-36)X \quad =-141.75$$

$$X^2+(2p-36)X+\frac{(2p-36)^2}{4}=-141.75+\frac{(2p-36)^2}{4}$$

(33) $$X = -\frac{2p-36}{2} \pm \sqrt{\frac{567+4p-144p+1296}{4}}$$

These two values of X indicate the values of X for the four points of intersection when the parabola passes thru the circle. In order that the parabola be tangent to the circle there should be only one value of X; therefore, the quantity under the radical sign should equal zero, and then

(34) $$X = -\frac{2p-36}{2}$$

In order to find the value of p, equate the radical to zero and solve for p

(35) $$\sqrt{\frac{4p^2-144p+729}{4}} = 0$$

(36) $p = 18 \pm 11.91 = (29.91)$ or 6.09

The value 29.91 has no meaning in this problem and therefore the equation of the required parabola is

(37) $Y^2 = 12.18X$

The point of tangency is determined from equation (34)

$$X = -\frac{2 \times 6.09 - 36}{2} = 11.91$$

and by solving equation (37) for Y for a value of $X = 11.91$

$$Y = \pm\sqrt{12.18 \times 11.91} = \pm 12.04$$

As the only point of interest in this problem is the one below the X axis the minus sign will be used and the point of tangency of the parabola and the circle is determined as (11.91,-12.04). These curves are plotted in Fig. 8.

The length of the center line may now be obtained by calculus methods. The general expression for the length of an arc of a curve is

$$(38)\quad L = \int_a^b \sqrt{1+\left(\frac{dx}{dy}\right)^2}\,dy$$

$$\text{or}\quad L = \int_a^b \sqrt{1+\left(\frac{dy}{dx}\right)^2}\,dx$$

Applying the formula to this problem the following is obtained:

$$(39)\quad L = \int_0^{-12.04} \sqrt{1+\left(\frac{y^2}{37.0881}\right)^2}\,dy$$

Performing the indicated operations

$$L_1 = \int_0^{-12.04} \frac{1}{6.09}\,(37.0881+y^2)^{1/2}\,dy$$

$$= \frac{1}{2x6.09}\left(y\sqrt{37.0881+y^2} + 37.0881\log(y+\sqrt{37.0881+y^2})\right)\Big|_0^{-12.04}$$

$$= \frac{1}{12.18}((162.25)+(37.0881x\ (\log(25.52-\log 6.09))))$$

L_1 = 17.68 ft., length of arc of the parabola. The circular arc may be determined by measuring the included angle thus

$$\alpha = \sin^{-1}\left(\frac{6.09}{13.50}\right) = 26^\circ 48' 54''$$

= 96534 seconds of arc. A circumference is equal to 1296000 seconds of arc. Thus the length of arc is equal to

$$L_2 = \frac{96534}{1296000} x 3.1416 x 13.5 x 2 = 6.32 \text{ ft.}$$

The length of the straight section may be obtained directly and is equal to 5.50 ft.

Thus the total length of the draft tube is equal to the sum of the lengths of these three sections or 29.58 ft.

The velocity of flow at any point in the draft tube must be related in a certain way to the velocities at other points. The velocoties at two points, however, are known at the outset; these are v_1 and v_n, the velocities at the top and at the end respectivly thus

$$v_1 = 0.25\sqrt{2gh} = 25.70 \text{ ft./sec.}$$

$$v_n = 4 \text{ ft./sec.} \quad \text{(assumed)}$$

v_n is governed by the tail race conditions, excavation, etc.

By means of these velocities the corresponding cross-sectional areas A_1 and A_n can be calculated, as the discharge Q is

known. The intermediate areas can be calculated only after the velocities through the planes in which these areas are located have been determined. The problem therefore is to find the velocity change in the draft tube per unit length measured along the center line. It must be remembered, however, that this change in velocity must be such that the change in velocity head becomes somewhat smaller than the change in static head. The idea is that one side of the film of water is subjected to pressure corresponding to the change in velocity head and the other side to a pressure corresponding to the change in static elevation. If the difference in pressures is such that the film is subjected to a back pressure the film will obviously expand sideways and thus fill the cross sectional area of the tube. Applying Bernoulli's theorem to this case,

$$(40) \quad \frac{V_a^2}{2g} + P_a + h_a = \left(\frac{V_{a+1}^2}{2g}\right) + P_{a+1} + h_{a+1}$$

but in accordance with assumption (b), on Page ,

$$(41) \quad \frac{v_a^2 \;\; v_{a+1}^2}{2g \;\;\; 2g} = h_{a+1} - h_a$$

so that $P_a = P_{a+1}$ or in other words the pressure remains constant.

If equation (41) is satisfied for the full load discharge through the draft tube, it is true that for smaller discharges

$$\left(\frac{v_a^2}{2g} \right) - \left(\frac{v_{a+1}^2}{2g} \right) < (h_{a+1} - h_a)$$

Consequently there will be a back pressure

$$(42) \quad P = (h_{a+1} - h_a) - \left(\frac{v_a^2}{2g} - \frac{v_{a+1}^2}{2g} \right)$$

In this formula h_a and h_{a+1} are the static elevations at the points a and a+1 on the center line while v_a and v_{a+1} are the corresponding velocities. From the condition that

$$(43) \quad v_a A_a = v_{a+1} A_{a+1} = Q$$

we obtain

(44) $v_a = \frac{A_{a+1}}{A_a} v_{a+1}$

and

(45) $v_a^2 = \left(\frac{A_{a+1}}{A_a} \right)^2 v_{a+1}^2$

Substituting this value of v_a^2 in equation (41) it follows that

(46) $\frac{v_{a+1}^2}{2g} \left(\frac{A_{a+1}}{A_a} \right)^2 - \frac{v_{a-1}^2}{2g} = h_{a+1} - h_a$

so that solving for v_{a+1}^2

(47) $v_{a+1}^2 = \frac{2g}{\left(\frac{A_{a+1}}{A_a} \right)^2 - 1} (h_{a+1} - h_a)$

This represents a parabolic equation and it proves that the velocity should be retarded in accordance with a curve cut from one of the branches of a parabola. In general the equation can be written

(48) $y^2 = KX$

However in this form it is not directly applicable to the problem. In order to make it applicable, it is necessary to shift the X-axis so that the apex of the curve will be

located on the Y-axis at a point with coordinates equal to 0, and b respectively.

Calling the new system of axes V and Z, any points on the curve that formerly had the coordinates X and Y will now have coordinates Z and V. Of these coordinates

(49) $Z = X$ and $Y + V = b$

as $Y = (b-V)$ it follows that

(50) $V = b \pm\sqrt{KZ}$

As in this case the lower branch of the curve only is used the equation finally becomes

(51) $V = b-\sqrt{KZ}$

Three different possibilities now appear, namely;

(1) $v_1+v_n > y_1+y_n$

(2) $v_1+v_n = y_1+y_n$

(3) $v_1+v_n < y_1+y_n$

In case (1) there is obvious danger that

change in velocity may become too great in the beginning of the tube, provided the difference is considerable. In case (3) it is apparant that if the difference is considerable the velocity curve becomes practically a straight line which is not desirable. Consequently only case (2) needs further consideration.

Since $y + v = b$, it follows that

(52) $y_1+v_1=b$ and

(53) $y_n+v_n=b$ or

(54) $y_1+v_1= y_n+v_n$

but from case (2) $v_1+v_n= y_1+y_n$

from which it follows that

(55) $2v_1 = 2y_n$

Thus, as $v_1 = y_n$ and $v_n = y_1$, it follows that

(56) $b = v_1 + v_n$

Substituting the coordinates Z_1 and v_1, and Z_n and v_n in equation (50), it becomes

(57) $v_1 = b - \sqrt{KZ_1}$ and

(58) $v_n = b - \sqrt{KZ_n}$

or, transposing,

(59) $b - v_1 = \sqrt{KZ_1}$ and

(60) $Z_1 = \frac{(b - v_1)^2}{K}$

In the same way it is proved that

(61) $Z_n = \frac{(b - v_n)^2}{K}$

But from equation (56), $b = (v_1 - v_n)$, it follows that

(62) $Z_1 = \frac{v_n^2}{K}$ and $Z_n = \frac{v_1^2}{K}$

(63) $L = Z_n - Z_1$

Therefore

(64) $L = \frac{v_1^2}{K} - \frac{v_n^2}{K} = \frac{1}{K}(v_1^2 - v_n^2)$

which gives the following value for K,

(65) $K = \frac{(v_1^2 - v_n^2)}{L}$

Introducing the values found in equations (56) and (65) in equation (51), it becomes

(66) $v = (v_1 + v_n) - \sqrt{\frac{v_1^2 - v_n^2}{L} Z}$

This is a convenient expression giving directly the valocity at any point Z on the center line. As previously mentioned v_1, v_n, and L have already been determined and can be considered constant quantities.

It is now required to locate the ordinates v_1 and v_n on the Z axis. Substituting in equation (66) the ordinates Z_1 and v_1 it becomes

$$(67)\quad v_1 = (v_1 + v_n) - \sqrt{\frac{v_1^2 - v_n^2}{L} Z_1}$$

From which

$$(68)\quad v_n = \sqrt{\frac{v_1^2 - v_n^2}{L} Z_1}$$

or solving for Z_1

$$(69)\quad Z_1 = \frac{L}{\left(\frac{v_1}{v_n}\right)^2 - 1}$$

and then

$$(70)\quad Z_n = L + Z_1$$

These equations will now be applied to

this problem in order to determine the velocities at various points along the center line. Applying equations (69) and (70), remembering that the length of the draft tube has been found to be 29.58 ft.,

$$Z_1 = \frac{29.58}{\left(\frac{25.7}{4}\right)^2 - 1} = 0.732 \text{ ft.}$$

and

$$Z_n = 29.58 + 0.732 = 30.31 \text{ ft.}$$

Substituting the velocities v_1 and v_n in equation (66) gives the velocity equation for this problem as

$$V = 30 - 4.72 \sqrt{Z}$$

This curve is shown in Fig. 9, from which it is possible to obtain the velocity at any point along the center line of the draft tube.

Points are taken along the center line and numbered as shown in Fig. 8. Then the velocities are obtained for these points and

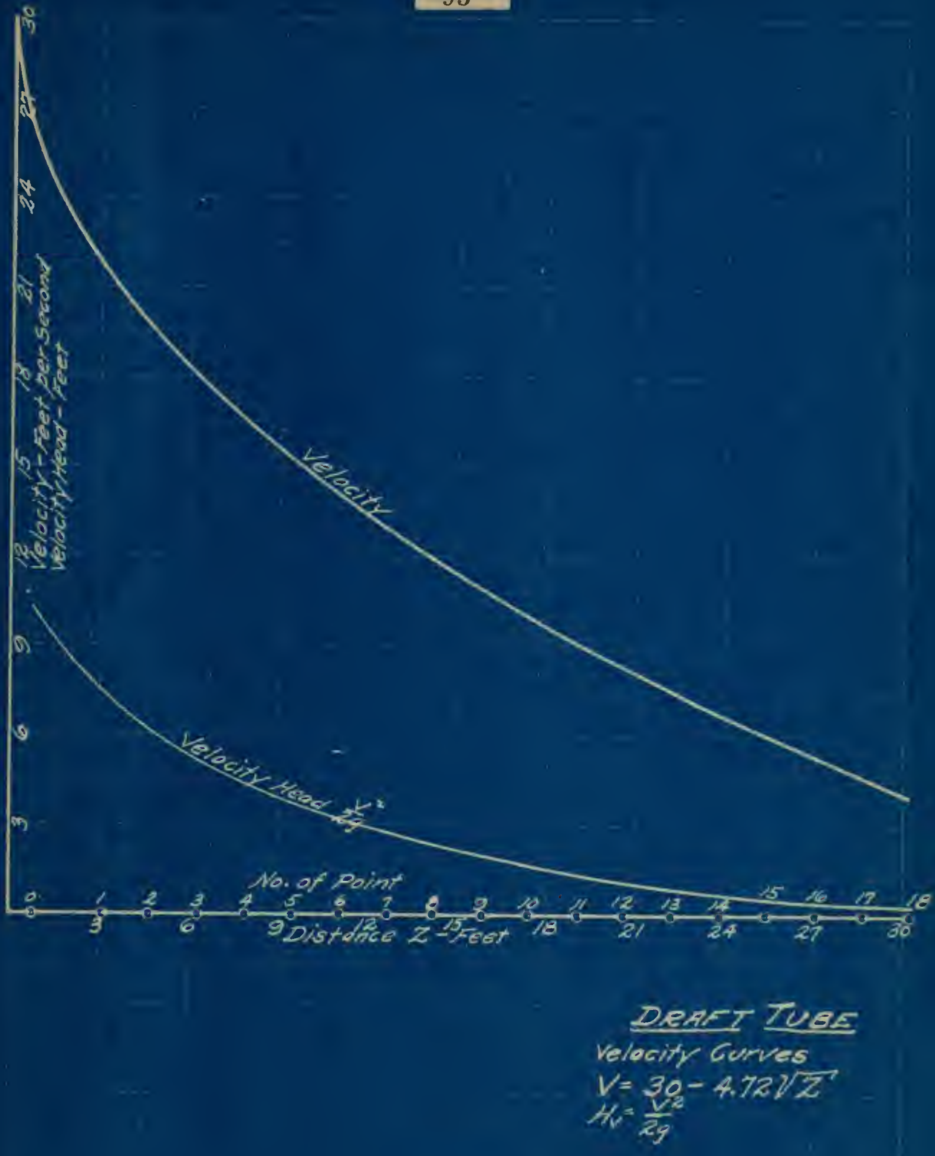

30
27
24
21
18
15
12
9
6
3
Velocity - Feet per Second
Velocity Head - Feet
Velocity
Velocity Head $\frac{V^2}{2g}$
No. of Point
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
3
6
9
12
15
18
21
24
27
30
Distance Z - Feet
DRAFT TUBE
Velocity Curves
$V = 30 - 4.72\sqrt{Z}$
$H_v = \frac{V^2}{2g}$

Fig. 9.

knowing the quantity of water Q, it is an easy matter to calculate the required cross-sections at these points. In Table 5 these values are shown with the changes in velocity head and static head between points. It can be seen that at only one point is the change in velocity head in excess of the change in static head, and this is in that part of the draft tube immediately below the the top of the runner band. As the exact velocities at this point are difficult to determine, and as the discrepancy is small it will be ignored.

The end section of the draft tube will be in the form of two equal semi-circles connected by tangents and having a major axis of 15' - 0", in order to allow 5 ft., of concrete between two adjacent draft tubes. In order to have 75 sq. ft., of area, the radii of these semi-circles must equal 2' - 8.5", making the minor axis of the section 5' - 5". The purpose in making these

Table 5.

DRAFT TUBE

NO. OF POINT	DISTANCE TO TOP OF TUBE	Z	V	$\frac{V^2}{2g}$	DIFF.	H	DIFF.	AREA OF SECTION
	FEET	FEET	FT/SEC.	FEET	FEET	FEET	FEET	FEET^2
0	0.00	0.72	25.70	10.23	2.73	19.08	2.41	11.55
1	2.44	3.16	21.80	7.40	1.22	16.67	1.59	13.85
2	4.07	4.79	19.95	6.18	1.10	15.08	1.58	15.05
3	5.70	6.42	18.10	5.08	.75	13.50	1.58	16.60
4	7.28	8.00	16.70	4.33	.58	11.92	1.50	17.98
5	8.86	9.58	15.52	3.75	.57	10.42	1.34	19.36
6	10.47	11.19	14.30	3.18	.48	9.08	1.26	21.00
7	12.11	12.83	13.20	2.70	.41	7.82	1.20	22.74
8	13.70	14.42	12.15	2.29	.34	6.62	1.00	24.72
9	15.28	16.00	11.20	1.95	.30	5.62	.92	26.80
10	16.88	17.60	10.30	1.65	.28	4.70	.91	29.16
11	18.49	19.21	9.40	1.37	.22	3.79	.79	31.90
12	20.11	20.83	8.60	1.15	.21	3.00	.79	34.90
13	21.70	22.42	7.78	.94	.21	2.21	.71	38.60
14	23.28	24.00	6.85	.73	.15	1.50	.67	43.60
15	24.88	25.60	6.20	.58	.13	0.83	.45	48.40
16	26.49	27.21	5.40	.45	.10	0.38	.28	55.65
17	28.11	28.83	4.79	.35	.10	0.10	.10	62.70
18	29.67	30.39	4.00	.25		0.00		75.00

end section other than circular is to reduce the amount of excavation required, and this, of course, is accomplished by making the major axis horizontal. As the cross-section decreases, the length of the connecting tangents will be decreased until they reach zero, and from this point to the top of the draft tube the reduction in area will be accomplished by reducing the diameter. It happens that the draft tube is circular from the top to about point 8, Fig. 8, and from this point to the discharge end it is composed of two semi-circles of radius 2' - 8.5" connected by tangents of length depending upon the required cross sectional area. The length of these tangents for any given required cross-sectional area is obtained from the formula which can be derived from the geometric relations of the quantities involved.

The passage from the intakes to the

discharge into the tail race has now been completely discussed, and there remains only the superstructure of the power house and the electrical equipment. The drawings number show the form of the power house and the arrangment of the equipment. A discussion of the size and controll of the electrical equipment will be given in Part 3.

PART 3.

ELECTRICAL EQUIPMENT.

E. H. SMITH.

The number of units to be

ELECTRICAL EQUIPMENT.

The power generated in this plant will be furnished by 3-3700 K.V.A., 8 pole, 6600 volt A.C., generators, the windings of which are star connected with the nuetral grounded. They will run at 375 r.p.m., and have a frequency of 25 cycles. Each of these machines will have an exciter mounted upon it's shaft to furnish the necessary direct current for field excitation. To provide against a breakdown of any one of these exciters, a small motor driven direct current generator will furnish power to busses which may be connected to the generators, after the self driven exciter is disconnected. The motor driven direct current generator will also furnish power for the control and operation of all switches, field control and water wheel control. To provide a method of starting the plant and also as a safeguard against the breakdown of the two methods of

excitation, a battery is installed which may be connected to the direct current busses when necessary.

The station will have 10 - 1400 K.V.A. single phase transformers, which step the voltage from 6600 volts up to 60,000 volts. Nine of these will be in use, 3 for each generator, and one will be kept as a spare unit. Both primary and secondary of the transformers are connected in delta. The reason for using 3 single phase transformers instead of 1 three phase transformer is on account of the cost of the latter type, especially as one extra set must be kept in readiness for a breakdown. In such a small plant as this the cost mentioned above for the extra unit would be entirely too large a percentage of the total cost.

On the two outgoing lines there are installed choke coils, horn gaps and lightening arresters, one in each conductor.

After leaving the generator the current passes through potential and current transformers which operate the ammeters, voltmeters, wattmeters, power factor meters, synchroscope and frequancy meter. Next is the machine switch of the K2 non-automatic oil type. Following this comes a K2 automatic oil switch to throw the generators on the low tension busses. To connect the machines to the transformers a K2 non-automatic oil switch is used. The transformers are connected to the high tension busses by means of an H6 automatic oil switch. There are two sets of outgoing lines, each one being connected to the high tension busses by means of an H6 automatic oil switch. The reason non-automatic switches were used on the generator side of the transformers was in case the machines were not in exact synchronism when the switch was thrown in. In this situation the sudden rush of

synchronizing current would throw open an automatic switch. All switches are distante controlled by 110 volt control circuits, operated by switches with individual signal lamps.

The field theostat of the main generator is motor operated and distant controlled. The switching from one system of excitation to another is done by two electrically inter-locked circuit breakers, distant controlled. A small shunt resistance is connected in the field circuit in case the field becomes grounded to the high tension circuit. This resistance will protect the field of the machine in such a situation.

The motor driving the direct current generator is started by 3 inter-locked switches, 2 of them connecting the motor to the busses through a compensator and the third connecting the motor to the line as it comes up to speed. The field rheostat for

this generator is a paul and ratchet type, distant controlled. The switch throwing the generator on the line is distant controlled. The switch which throws the battery on the line is also distant controlled, but is operated by battery current only, so that it can be thrown even though there is no power in the direct current busses.

On the exciter switchboard panel are four control switches; one for starting the driving motor, one for operating the generator switch, one for operating the field rheostat and one for the battery control. On the meter board are two ammeters, one for the generator and one for the motor, and also one voltmeter, used as a ground detector and for the generator. The voltmeter is operated by plugs.

On the main generator panel are 9 controll switches; they operate the machine, low tension bus, transformer, high tension

bus, ground and exciter switches, and the field rheostat and water wheel control. On the meter board are three a.c., ammeters and one direct current field ammeter, a power factor meter, a recording wattmeter and an integrating wattmeter. The integrating wattmeter is placed on the back of the board, as it is only necessary to inspect it once in a while. On the end of the board are placed the synchroscope, frequency meter, voltmeter and ground ammeter. The voltmeter and frequency meter are operated by one set of plugs and may be connected into any phase of the line; the synchroscope is operated by another set of plugs.

On the high tension panel are two control switches for the two H6 outgoing switches. On the meter board are placed six ammeters, one for each conductor in each of the two lines.

The H6 switches and the horn gaps are

arranged with disconnects on each side of them to provide for any repairs or alterations which may be necessary.

The generators are protected by the Merz-Price system for overload. This means that if an overload occurs in either the ground leads or the power leads the machine switch and both field switches will be opened automatically. The same system is used on the transformers and will open the K2 and H6 switch on either side of the transformers thus throwing them off the line.

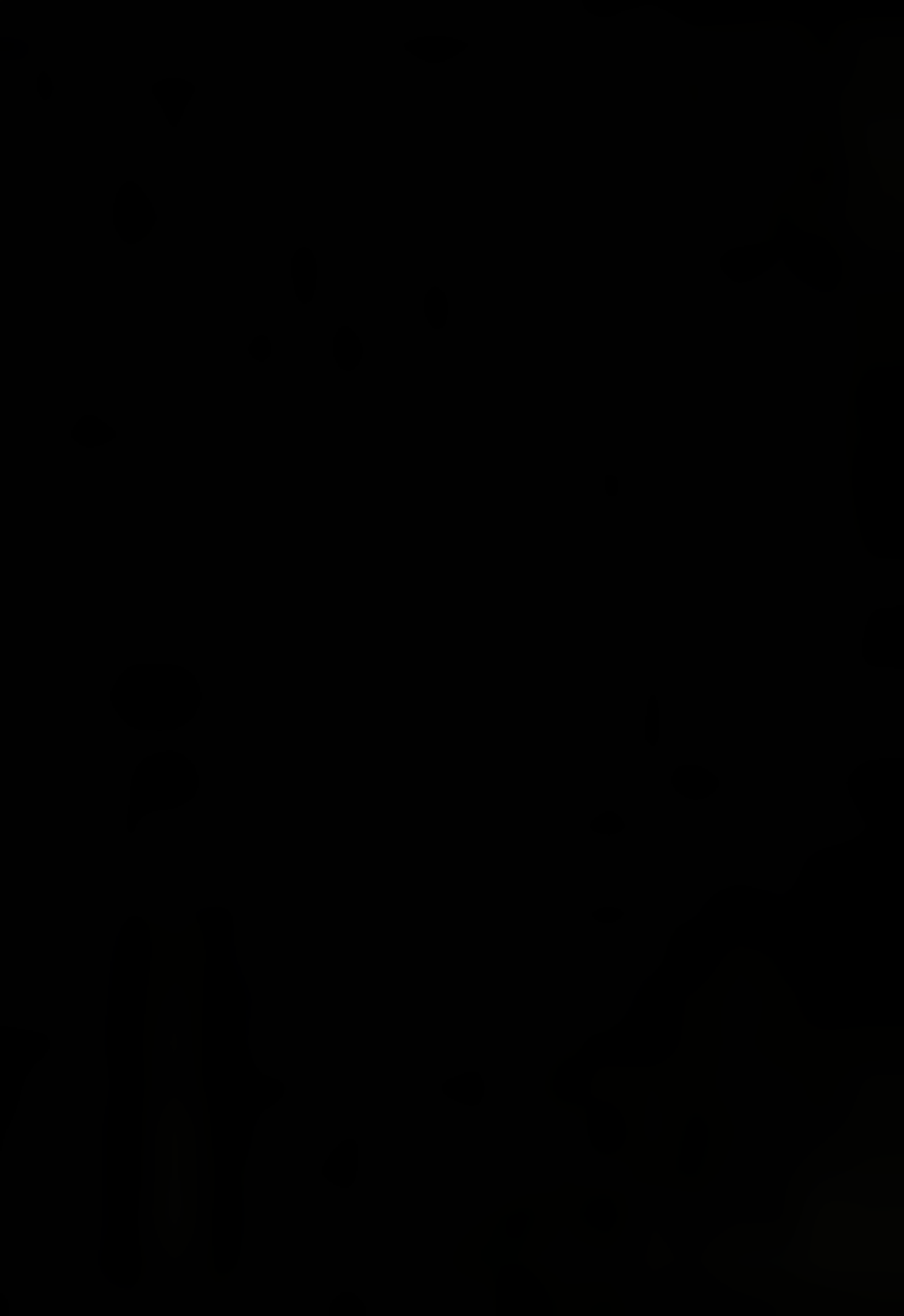
The high tension busses are grounded through grounding transformers, whose primary is star connected with the middle point grounded, and whose secondaries are delta connected with a resistance in circuit. The busses also have frequency absorbers and a ground detector connected to them.

The entire electric system is distant controlled by switches provided with signal

lamps and is automatically protected against any overload. In fact every safeguard which is known is used to protect the system and machinery.

To aid the operator in bringing the frequency to the right value a switch is placed on the generator panels for the control of the water wheel gates, by means of a small electric motor. Thus the operator may adjust the frequency without moving from the board.

The switchboard is of the bench type, with all control switches and plug receptacles located on the inclined bench. Resting upon the back of the bench is the meter board. There are five panels, one for the exciter, three for the generators and one for the feeders. The entire board is constructed of channels and iron pipe, thus affording a very rigid construction.

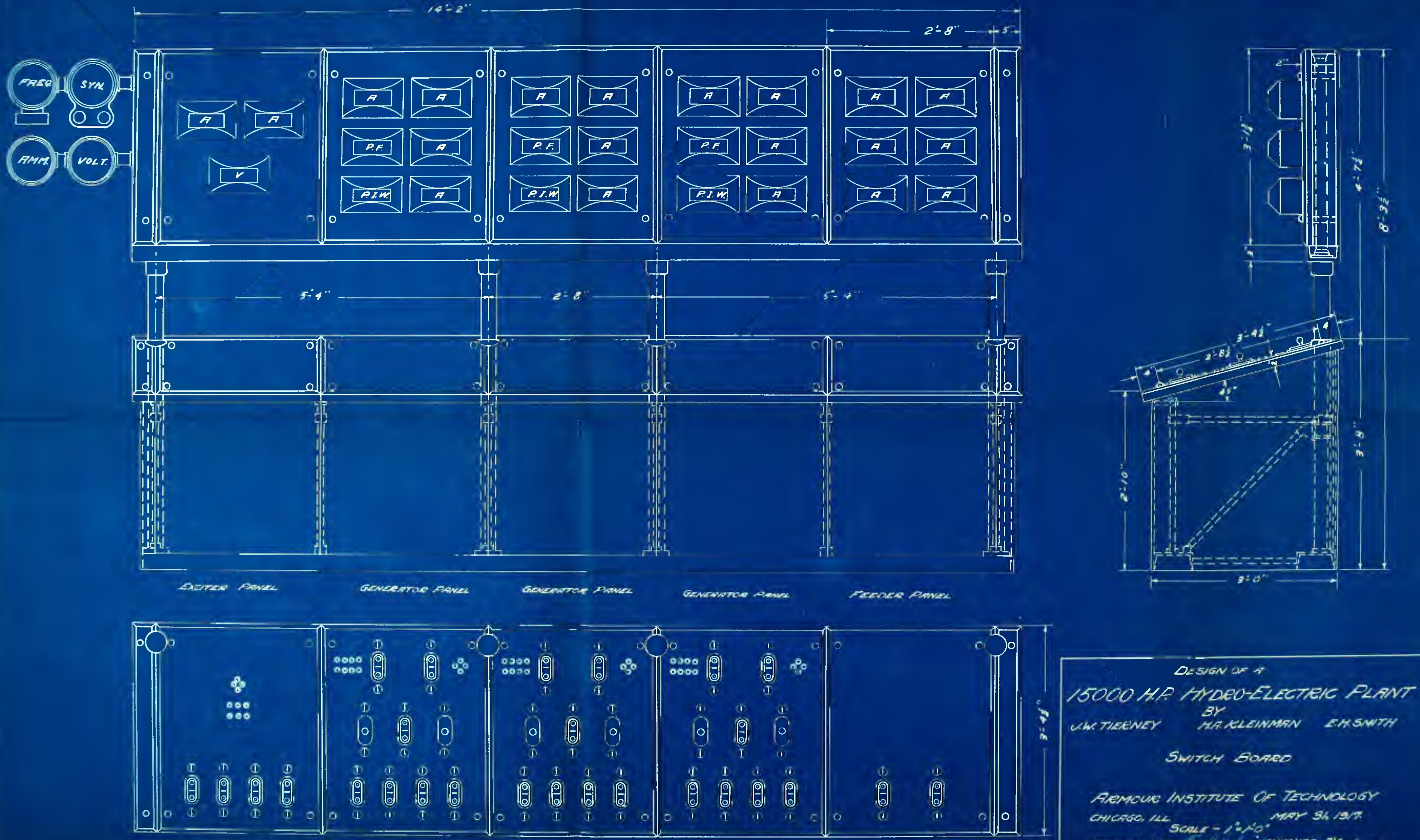


SANITARY SEWERAGE SYSTEM FOR THE VILLAGE OF CRETE ILL.

PROFILE PAGE	STREET	HIGH		LOW		FALL IN FEET	LENGTH IN FEET	PER CENT GRADE	DIAMETER IN INCHES	KIND OF PIPE	AVERAGE DEPTH IN FEET	WIDTH OF TRENCH IN INCHES	EXCAVAT. IN CU. YDS.	LENGTHS OF PIPE IN FEET	CUSEC. DRAINING INTO	CUSEC. FLOWING OUT	MAN-HOLES	DROP M'HOLES	MISCELL-ANEOUS	Y CONNECTION
		FROM	ELEV.	TO	ELEV.															
9	SECOND ST.	100'E. OF PEORIA AVE.	90.40	COLUMBIA ST.	85.40	5.0	1325	.38	6	V	11.80	24"	555	1335	.059	.059	3			44
8	FIRST ST.	50' " " " "	95.60	" "	84.60	11.0	1378	.80	"	"	10.80	"	600	1430	.059	.059	3			44
"	JEFFERSON ST.	" " " " "	92.60	LINCOLN ST.	87.60	5.0	850	.59	"	"	10.65	"	336	860	.044	.044	2			32
"	LINCOLN ST.	250' S. OF FIRST ST.	89.00	EXCHANGE ST.	84.20	4.8	480	1.00	"	"	9.40	"	169	510	.011	.055	2			8
3	COLUMBIA ST.	FOURTH ST.	87.00	" "	82.60	4.4	1580	.28	"	"	8.20	"	967	1590	.057	.175	5			42
10	DIVISION ST.	350'E. OF COLUMBIA ST.	87.20	COLUMBIA ST.	85.40	1.8	350	.51	"	"	10.05	"	273	355	.014	.014	1			10
3	COLUMBIA ST.	500'N. OF FARM ST.	93.60	EXCHANGE ST.	82.40	11.2	2220	.50	"	"	10.95	"	1830	2250	.108	.122	5			70
9	LUMBER ST.	FOURTH ST.	83.60	FIRST ST.	81.00	2.6	910	.29	"	"	9.10	"	617	915	.034	.034	3		◒	25
8	FIRST ST.	LUMBER ST.	80.00	MAIN ST.	77.80	2.2	730	.30	"	"	7.50	"	405	730	.013	.047	2			10
4	BENTON ST.	50'N. OF NORTH ST.	81.90	EXCHANGE ST.	80.10	1.8	450	.40	"	"	11.25	"	380	455	.019	.019	1			14
"	" "	DIVISION ST.	84.70	" "	80.10	4.6	785	.59	"	"	14.30	"	647	800	.035	.035	2			26
2	EXCHANGE ST. WEST	PEORIA AVE.	86.90	COLUMBIA ST. NORTH	82.60	4.3	1430	.30	"	"	6.65	"	755	1535	.059	.115	4			44
"	" " "	N. COLUMBIA ST.	82.58	MAIN ST.	79.00	3.6	1190	.30	8	"	15.10	"	1342	1200	.032	.498	3		◓	24
"	" " EAST	EAST ST.	83.20	ELIZABETH ST.	80.40	2.8	1360	.21	6	"	9.20	"	930	1365	.058	.120	4			46
"	" " "	WOOD ST.	88.50	" "	85.00	3.5	360	.98	"	"	9.60	"	270	380	.011	.036	1			8
11	WOOD ST.	100' S. OF NORTH ST.	89.35	EXCHANGE ST.	88.40	.95	215	.44	"	"	8.35	"	151	218	.011	.011	1			8
"	" "	100' N. OF CASS ST.	90.20	" "	88.40	1.8	440	.41	"	"	10.30	"	340	445	.014	.014	1			10
"	ELIZABETH ST.	60' " " " "	89.40	" "	86.00	3.4	525	.65	"	"	10.15	"	406	540	.014	.014	1			10
"	" "	EXCHANGE ST.	80.40	NORTH ST.	79.85	.55	275	.20	"	"	16.20	30 "	413	275	.008	.169		1		6
10	NORTH ST.	ELIZABETH ST.	79.80	MAIN ST.	78.30	1.5	720	.21	"	"	16.85	"	1135	728	.022	.191	2			16
5	CASS ST.	EAST ST.	86.20	" "	80.00	6.2	2030	.30	"	"	13.60	24"	2040	2035	.068	.068	6			50
"	DIVISION ST.	" AVE.	90.50	" "	86.60	3.9	1320	.30	"	"	8.70	"	860	1330	.041	.080	4			30
7	HERMAN ST.	" "	101.60	STA.10 530'W. OF EAST ST.	100.00	1.6	530	.30	"	"	8.20	"	325	535	.024	.078	2			18
"	" "	STATION 10	100.00	MAIN ST.	89.30	10.7	765	1.40	"	"	7.90	"	518	835	.035	.112	1			25
6	ENTERPRISE AVE.	70' N. OF X ST.	96.20	CRETE BLVD.	93.60	2.6	650	.40	"	"	8.60	"	420	660	.073	.073	1			54
7	HEWES AVE.	" " " " "	97.20	" "	94.60	2.6	650	.40	"	"	9.00	"	440	660	.073	.073	1			54
6	AUSTIN AVE.	" " " " "	98.40	" "	95.80	2.6	650	.40	"	"	9.25	"	452	660	.073	.073	1			54
"	EAST AVE.	" " " " "	99.00	" "	97.00	2.0	650	.31	"	"	9.35	"	457	660	.073	.073	1			54
"	" "	Y ST.	100.40	" "	97.00	3.4	550	.62	"	"	10.85	"	446	555	.054	.054	1			40
10	FAITHORN AVE.	" "	101.40	" "	98.20	3.2	500	.64	"	"	8.00	"	298	505	.054	.054	1			40
"	" "	X ST.	99.90	" "	98.20	1.7	500	.34	"	"	4.90	"	182	505	.054	.054	1			40
6	CRETE BLVD.	FAITHORN AVE.	98.20	MAIN ST.	92.40	5.8	1600	.36	8	"	11.35	"	1350	1610	.031	.384	5			24
4	LINDEN ST.	450' W. OF MAIN ST.	78.40	" "	76.10	2.3	450	.51	6	"	8.50	"	296	455	.019	.019	1			14
"	FIFTH ST.	420' " " " "	77.40	" "	75.40	2.0	420	.46	"	"	8.00	"	253	425	.019	.019	1			14
1	MAIN ST.	X ST.	94.64	CRETE BLVD.	92.40	2.2	745	.30	"	"	15.25	"	508	750	.073	.073	1			54
"	" "	CRETE BLVD.	92.40	HERMAN ST.	89.40	3.0	1000	.30	8	"	10.65	"	795	1010	.054	.576	2			40
"	" "	HERMAN ST.	89.30	CASS ST.	85.60	3.7	1225	.30	10	"	10.35	"	943	1230	.070	1.085	3			52
"	" "	CASS ST.	85.60	EXCHANGE ST.	84.20	1.4	470	.30	12	"	16.20	30"	735	490	.035	1.188	1			26
"	" "	EXCHANGE ST.	79.00	FIFTH ST.	75.40	3.6	1805	.20	14	"	10.55	"	1835	1810	.065	2.027	5	1	◑	48
"	" "	725'N. OF FIFTH ST.	77.00	" "	75.40	1.6	725	.22	6	"	9.22	24"	495	730	.032	.032	2			24
4	EXTENSION OF FIFTH ST. EAST	MAIN ST.	75.40	900 E. OF MAIN ST.	73.60	1.8	900	.20	14	"	6.26	"	434	905	.000	2.060	1		◐	0

◓ 2-12' LENGTHS OF 8" C.I. PIPE - WEIGHT 61.7# PER FOOT
◒ 2-12' " " 6" " " " 41.7# " "
◑ 2-12' " " 14" " " " 145# " "
} AT POINT WHERE LINE CROSSES CREEK. PIPE SUPPORTED BY STANDARD HANGERS FASTENED INTO WINGED-WALL ABUTMENTS

◐ 14-12' " " 14" " " " 145# " " WOODEN TRESTLE ABOUT 150' LONG AND 6' HIGH NEEDED.



DESIGN OF A
15000 H.P. HYDRO-ELECTRIC PLANT
BY
J.W. TIERNEY H.R. KLEINMAN E.H. SMITH

SWITCH BOARD

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SCALE – 1"=1'-0"
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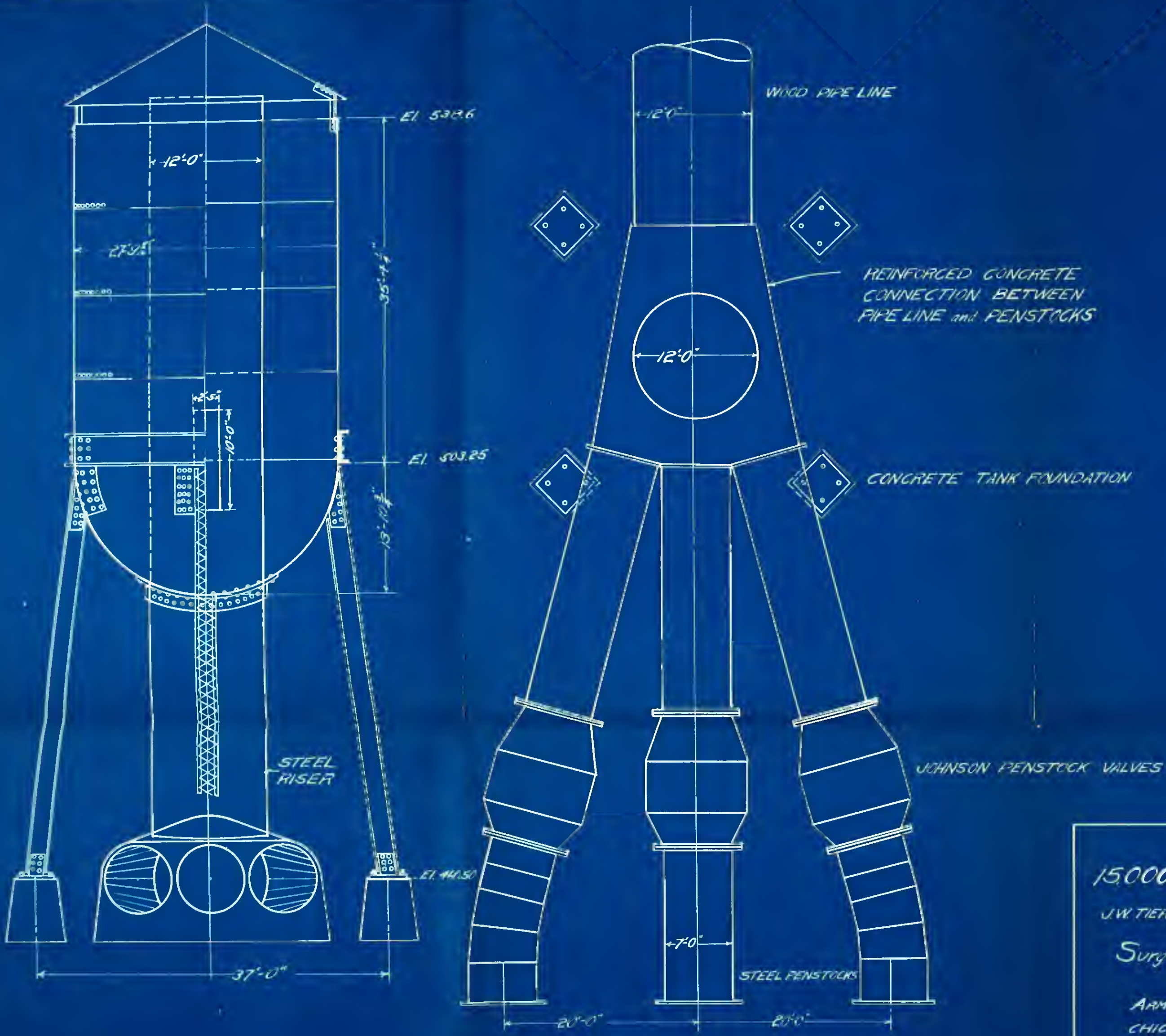

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Surge Tank – Penstock Connection
2

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SCALE – ⅛" = 1 FOOT
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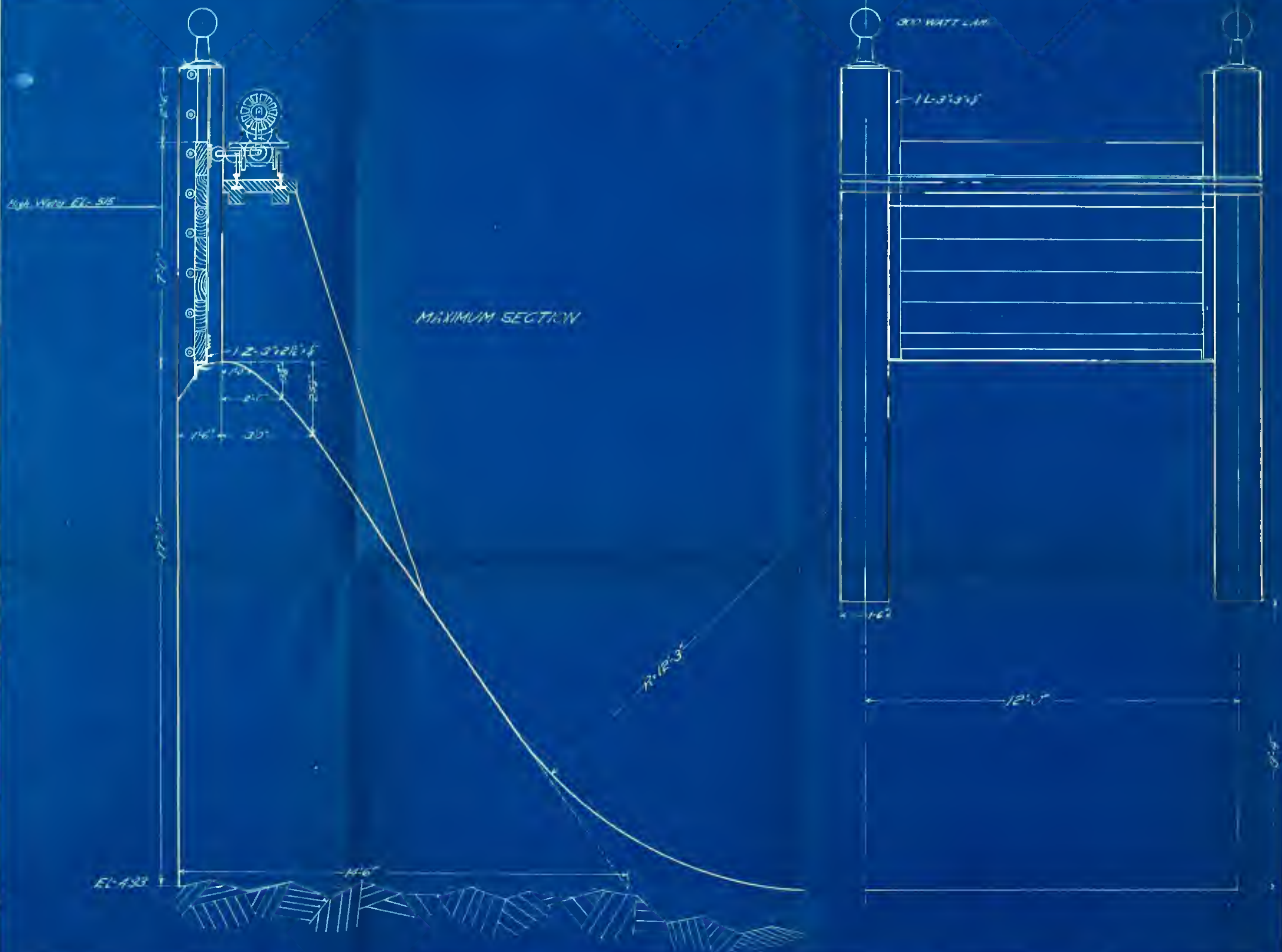

DESIGN OF A
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SPILLWAY SECTION
#1

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BY

J.W. TIERNEY H.A. KLEINMAN E.H. SMITH

Dam Section and Topography

3

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SCALE - AS SHOWN

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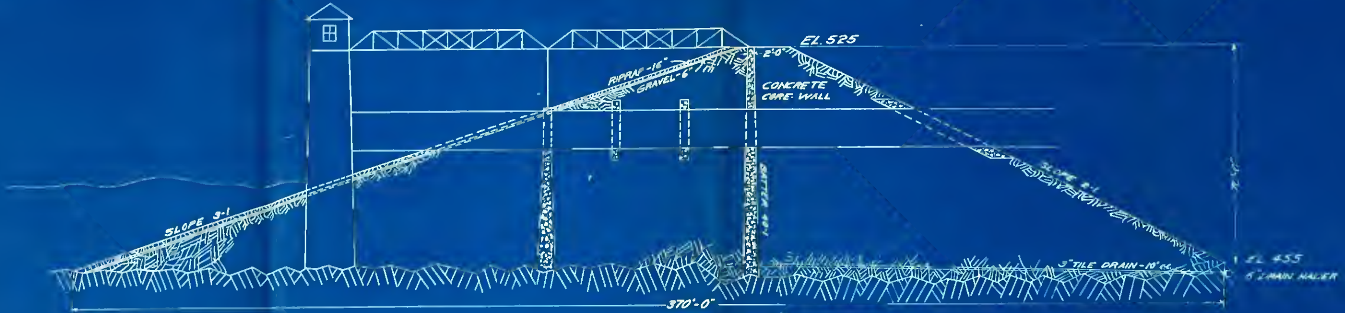
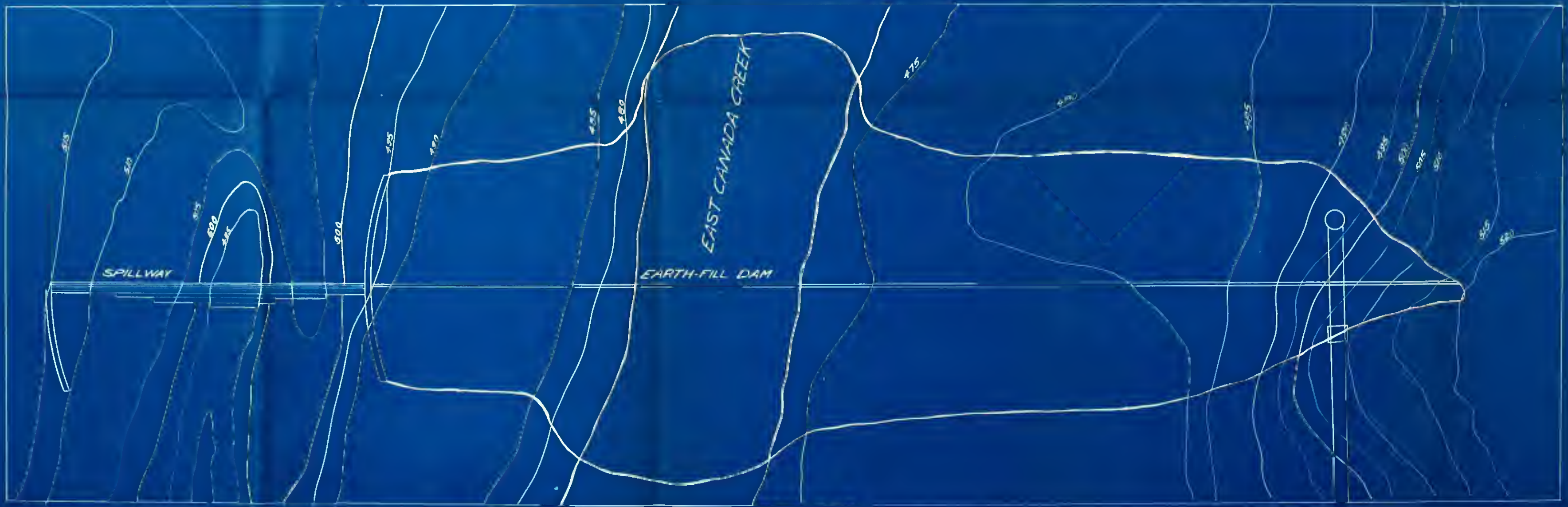
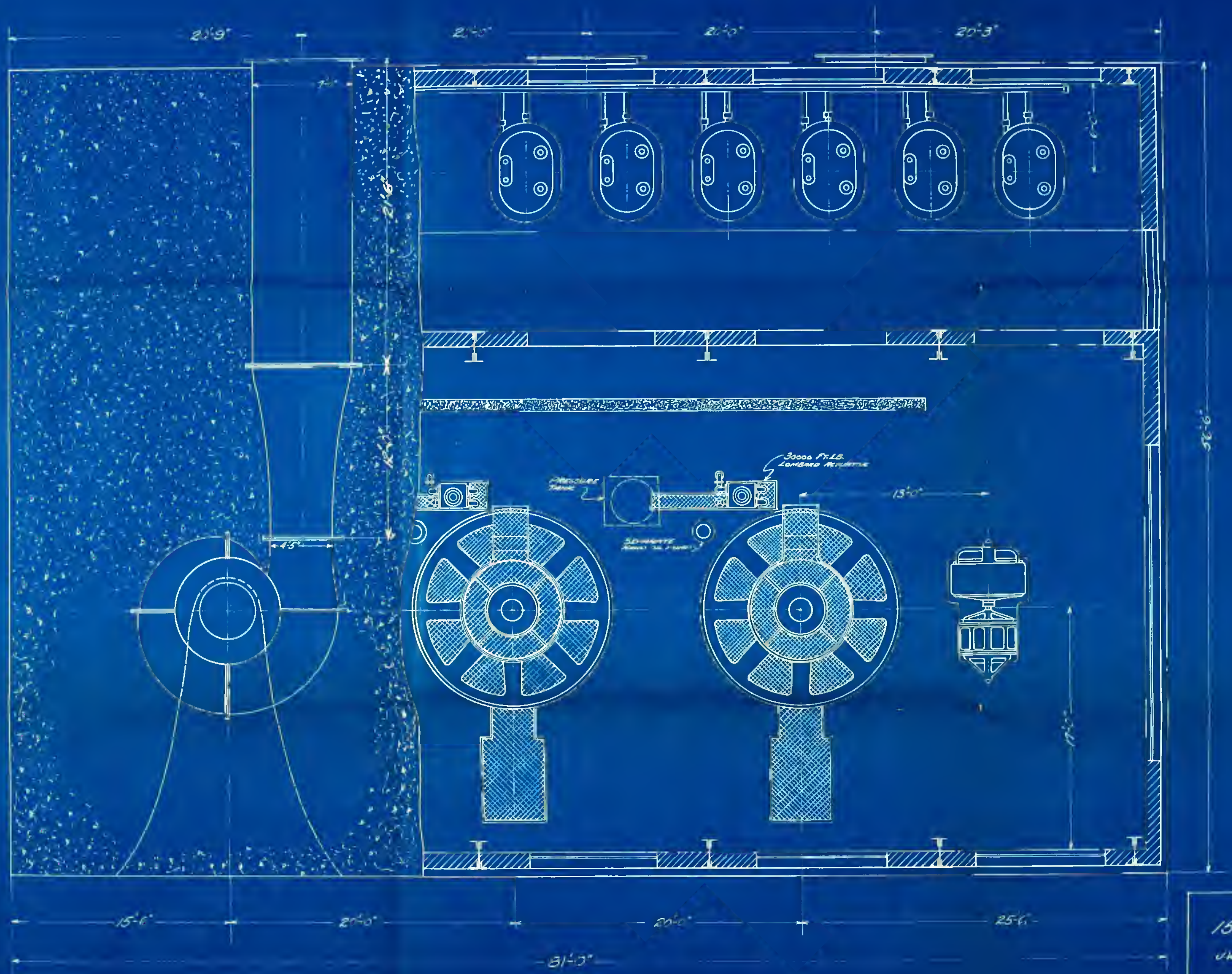


SCALE-1"=20'



SCALE-1"=100'



DESIGN OF A

15000 H.P. HYDRO-ELECTRIC PLANT

BY

J. W. TIERNEY   H. R. KLEINMAN   E. H. SMITH

PLAN OF POWER HOUSE

ARMOUR INSTITUTE OF TECHNOLOGY

CHICAGO, ILL.   MAY 31, 1917.

SCALE – 1/8" = 1'-0"

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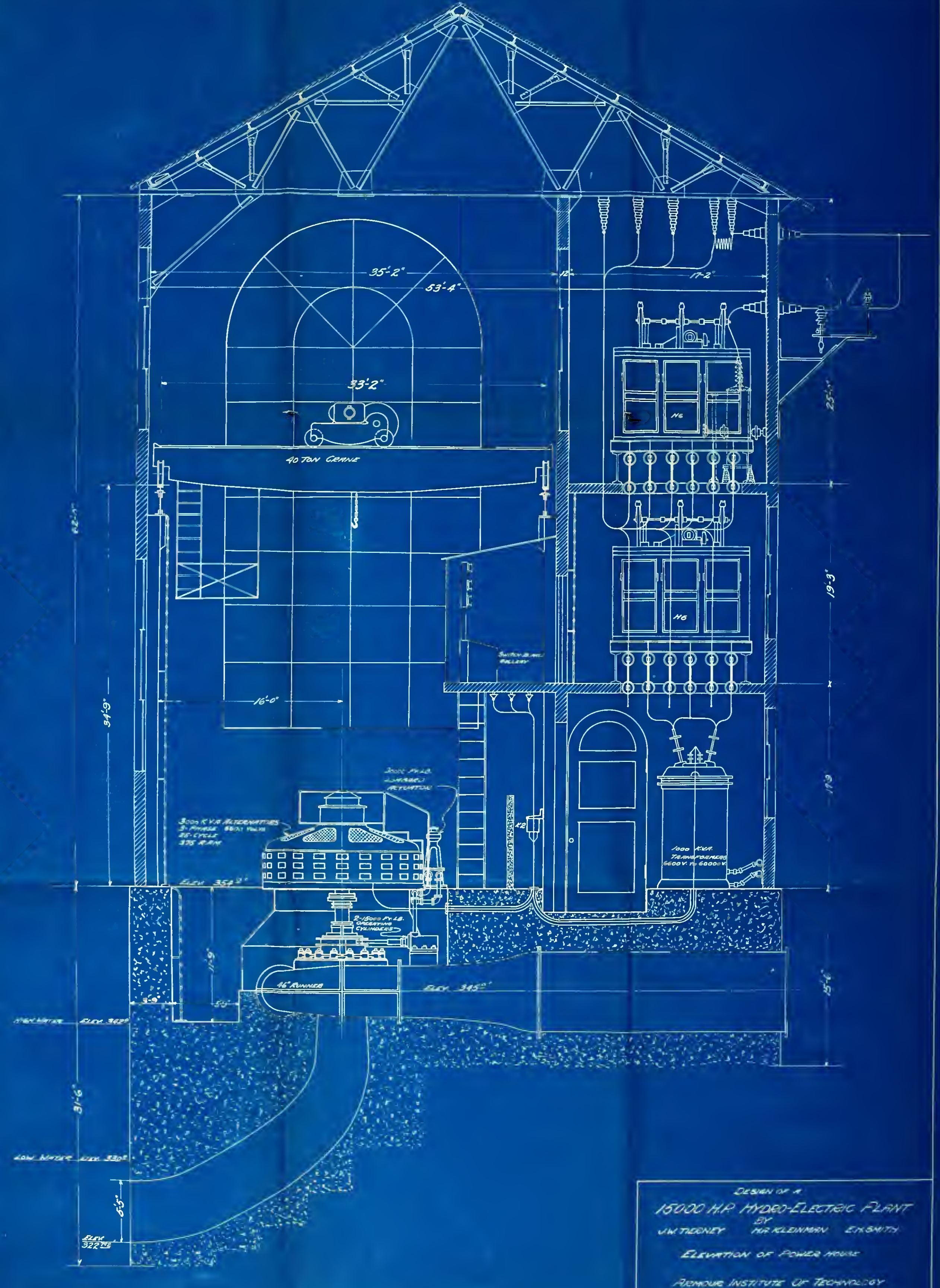


Design of a

15000 H.P. Hydro-Electric Plant

by

J.W. Tierney H.A. Kleinman E.H. Smith

Elevation of Power House

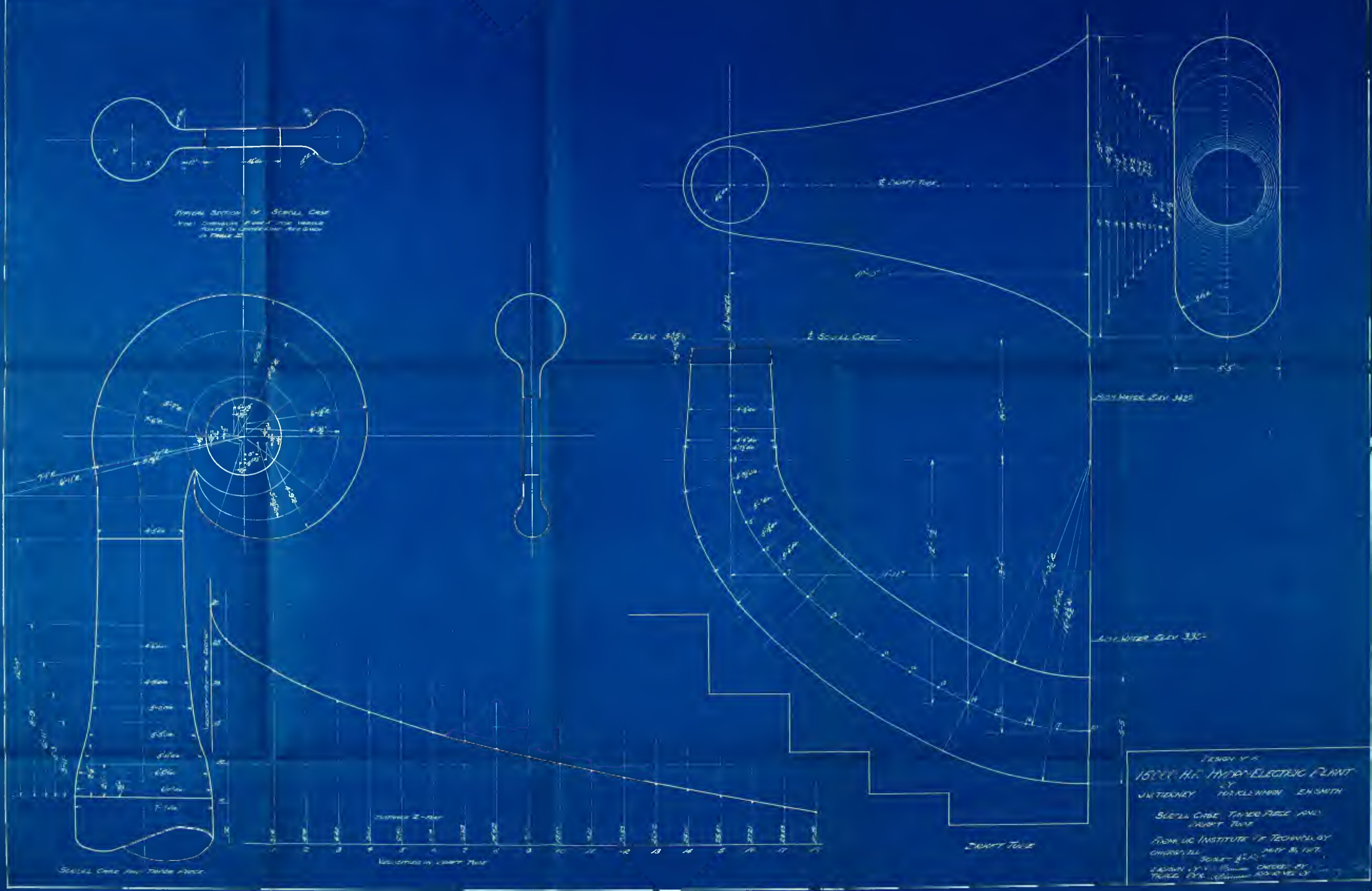
Armour Institute of Technology

Chicago, Ill. May 31, 1917

Scale – 3/8" = 1'-0"

Drawn by H.A. Kleinman Checked by

Traced by H.A. Kleinman Approved by


TYPICAL SECTION OF SCROLL CASE
NOTE: DIMENSIONS R AND r FOR VARIOUS POINTS ON CENTER-LINE ARE GIVEN IN TABLE II
SCROLL CASE AND TAPER PIECE
VELOCITIES IN DRAFT TUBE
DISTANCE Z - FEET
ELEV. 343.5
℄ SCROLL CASE
℄ DRAFT TUBE
HIGH WATER ELEV. 342.5
LOW WATER ELEV. 330.0
DRAFT TUBE
5'-5"

DESIGN Nº 4

16000 H.P. HYDRO-ELECTRIC PLANT
BY
J.W. TIERNEY H.R. KLEINMAN E.H. SMITH

SCROLL CASE, TAPER PIECE AND
DRAFT TUBE

ARMOUR INSTITUTE OF TECHNOLOGY
CHICAGO, ILL. MAY 31, 1917.
SCALE — ¼" = 1'-0"
DRAWN BY ... CHECKED BY
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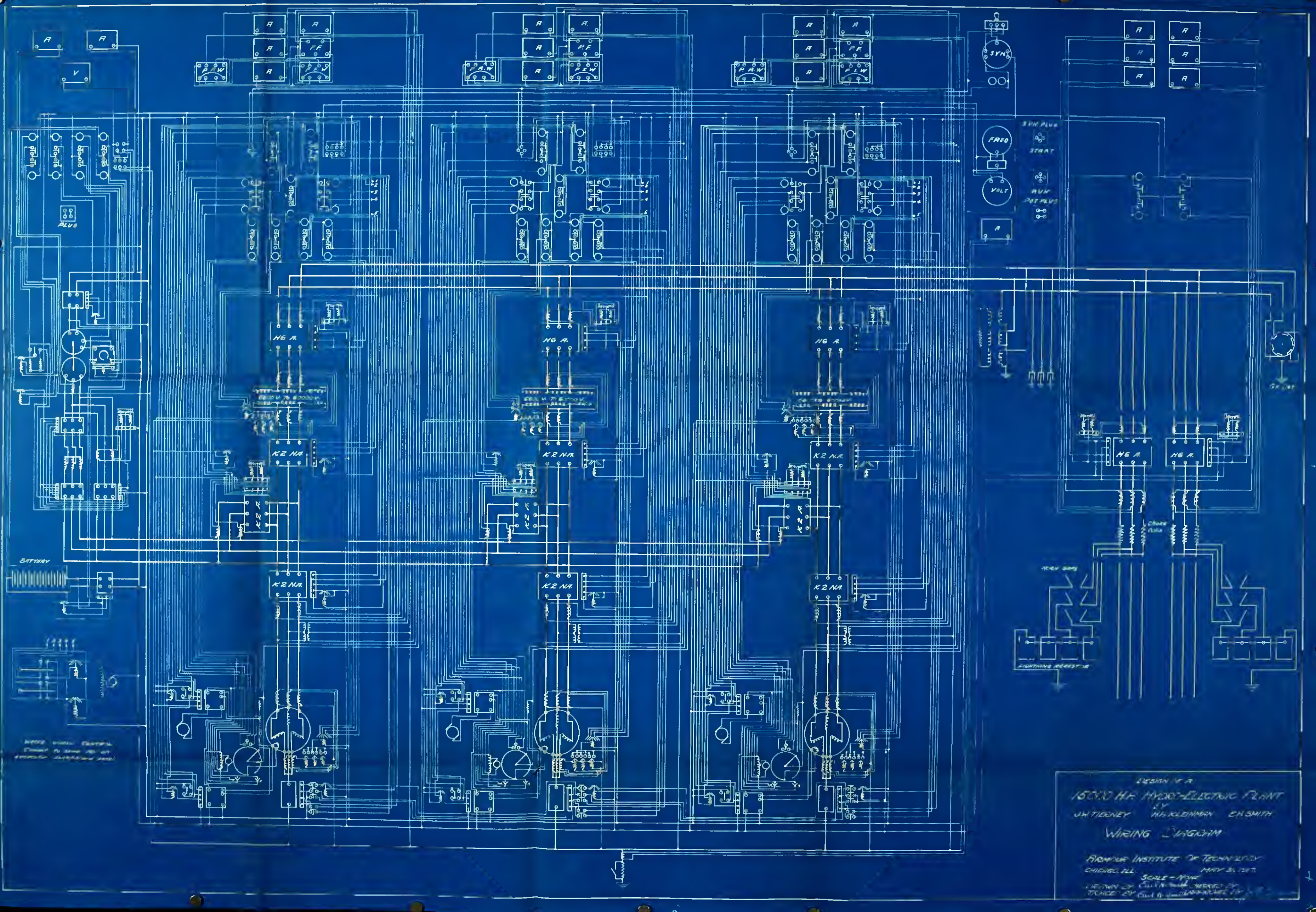

DESIGN OF A
15000 H.P. HYDRO-ELECTRIC PLANT
BY
J.H. TIERNEY M.H. KLEINMAN E.H. SMITH

WIRING DIAGRAM

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